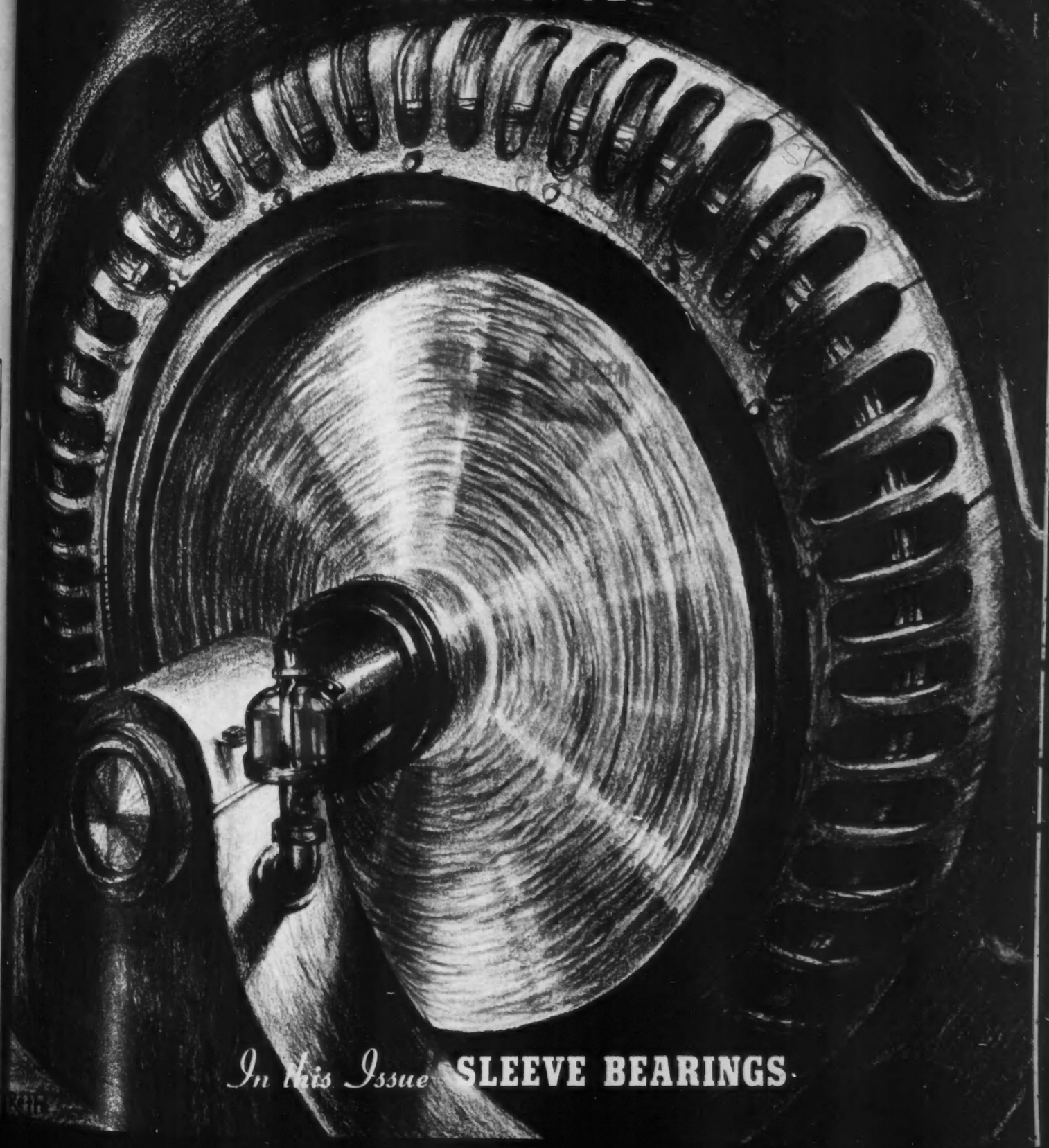
PRODUCT ENGINEERING

MAY, 1942



In this Issue **SLEEVE BEARINGS**

Conversion TO WAR PRODUCTION CALLS FOR MORE INFORMATION ABOUT ALLOYS...

Conversion to war production makes great demands upon both plants and personnel. While altering plant layouts, experienced employees must be taught correct methods of handling new operations on different metals. New employees must be trained...and taught to avoid waste and spoilage of critical materials.

You can *quickly* obtain practical answers to questions about the selection, fabrication and uses of ferrous and non-ferrous alloys containing Nickel by asking us. We have on hand a fund of information collected through years of research, field studies and experiences of alloy users.

This data has been checked and edited into convenient charts and pamphlets. These printed pieces range from technical data for engineers to simplified guides for apprentices.

Now...with minutes and materials so vital to Victory...make full use of this metal-working experience. Our technical staff also offers personal assistance in overcoming problems created when Nickel was allotted to places where it serves the Nation best.

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PRODUCT ENGINEERING

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IN THIS ISSUE

Sleeve Bearings

Before he can design a bearing with assurances that it will do its job satisfactorily, the product engineer must understand how to coordinate the interrelated design factors that determine a bearing's performance. Bruno Sachs, technical director of Johnson Bronze Company, explains these factors and how to use them in this month's lead article, first of a comprehensive, data-packed series. Part I, pages 248 to 252, describes heat generation, effect of lubricant, friction states. Charts show relations between bearing diameter, length, clearance, thickness of oil film, speed, and unit bearing pressures.

Industrial Plastic Parts

Industrial plastics have a man-sized job to do in tough mechanical, electrical and chemical applications. In spite of the large volume used in saltcellars, toys, molding and fancy gadgets that the average man may think of when you say "plastics," industrial applications account for the greatest volume of plastics production. Conclusion: every plastic part ought to be designed with the same care as a spring, casting, or bearing. That means the design engineer should know thoroughly everything contained in T. N. Willcox's article "Industrial Plastic Parts," pages 269 to 272.

Fits and Tolerances

An engineer called our office the other day and said he was checking over a blueprint which, he suspected, called for the wrong class of fit for a shaft. Could we help him?, he asked. It seems the American Standards Association has recently set up tentative standards for fits and tolerances, but naturally these do not indicate what class should be specified for any given application. Our answer: when faced with this problem of fits and have little previous experience to go by, turn to pages 278-280 for 19 sketches of typi-

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cal applications suggesting what class of fit to try first.

For quick determination from a nomograph of tolerances, allowances and interferences for the eight classes of fit included in the American Tentative Standard B4a—1925 of the American Standards Association, turn to this month's Reference Book Sheet, page 305.

Welded Strap Ends

Strap ends for brake bands are common machine elements, but there is plenty of opportunity for improvements in their design by use of welding. In an article beginning on page 262, theory, design and savings in time, material and costs of welded strap ends are shown by Mr. Gustaf Johansson.

Patent Procedure—IV

Suppose you've had a good idea and want to develop it with assistance. Will you be able to prove later that it was all yours? Will you lose your right to a patent through ignorance of limitations on sales and public use before an application is filed? Mr. Miles Henninger, in another of his series tells you what to do and what to avoid in order to complete your invention satisfactorily. Pages 281-282.

Brightness Ratings

There are many times when information on the tarnish rate of a metallic surface is extremely valuable. For instance: the design of electrical circuits where metallic contacts are exposed to atmosphere for long periods. Alfred M. Suggs of P. R. Mallory & Company describes on page 253 of this issue a new method to analyze tarnish by measuring light absorption and shows how to set up the apparatus required.

Hot

Induction heating has been a "hot" subject in progressive design during the past very few years. We've seen it take over hitherto impossible jobs in surface hardening. We've seen it giving scale-free heats for precision forging. Just now it's moving into the field of brazing which, mark our words, will soon be mighty important. Brief data: page 261.

Selenium Rectifiers—II

PRODUCT ENGINEERING for March plunged into the new and important subject of selenium rectifiers with Part I of this comprehensive article by Carole A. Clarke. In this closing installment Mr. Clarke gives additional information and data on performance of selenium rectifiers and presents basic tables for the design of circuits using these rectifiers. Size and capacity of rectifiers, circuit characteristics and current factors of the different circuits are all included. An article for your reference files: pages 266 to 268.

Hydraulic Cylinders—II

Concluding article in this series by James E. Thompson and Rodney B. Campbell deals mainly with design details for sealing various types of hydraulic cylinders, such as end caps, swivel joints, and ring seals. Methods for limiting length of stroke are included, as are designs of dash-pots that serve as shock absorbers. Pages 274 to 277 in this issue.

Electron Tubes vs. Mechanisms

An industrial instrument takes the lead among this month's Modern Designs. It is a self-balancing potentiometer pyrometer — formerly an instrument requiring a galvanometer and complicated cyclic balancing mechanisms with all the design and service problems attendant. Brown Instrument Company have recently replaced this galvanometer with a continuous balancing system consisting only of an electronic unit and a simplified mechanical drive—an interesting electrical-mechanical design with precise, fast, and persistant operation as its aim. For details see pages 254 to 256.

Materials Data

Use of silver-bearing solders and brazing alloys is expected to expand because of restrictions on tin. This article by Herbert Chase covers properties of these alloys, their advantages, and their relation to proper design of joints. See pages 283 and 284.

Vinylidene chloride plastics are undoubtedly among the most important recent developments in materials. They are extremely tough, strong and chemical-resistant, and can be molded, extruded, or pulled into high-strength fibres. Complete design data is given on pages 285 and 286.

Neoprene synthetic rubber has a wide background of experience which constitutes an advantage over more recently introduced synthetic rubbers. Much has been published about neoprene's characteristics and behavior in service. Pages 287 to 289 abstract the more significant qualities.

Physical properties of Muntz metal, an alloy of copper and zinc which contains 60 percent copper, are given in chart form on pages 290 to 294. This alloy dates back to a patent by Sir Frederick Muntz in 1832.

WHAT'S COMING

Electric-Strain Gages

The strain gage opens the path for more rational and efficient product design based on a knowledge of actual service loads and conditions. Strain gage studies enable the designer to put strength where it is needed, and eliminate wasted material where strength is not needed. A relatively new and simple method for measuring strain in stressed materials now makes it possible for the design engineer to measure more easily and exactly the loads, stresses and strains induced in structures. Details of this new method will be given next month in "Electric-Strain Gages" by F. Mittelberger. A good article for all interested in design testing.

Which Molded Plastic?

When designing molded plastic parts, the choice between injection and compression molding compounds is often a tough one. Next month, D. A. Dearle, author of the book "*Plastics Molding*" will answer common questions on this subject. Most of these questions involve design of the molds and what you can do with them.

Universal Joints by Vectors

One of the bugaboos in designing a universal joint is that the variation in load is not apparent by a quick inspection, and without a good analysis some bad errors might creep in. Engineers L. D. Hagenbush and J. Holstein of Goodman Manufacturing Company have developed a vectorial approach to this problem that can save much time over the drawing board. They will explain it in PRODUCT ENGINEERING for June.

Gear Tooth Profiles

Gears 1 in. wide that carry 2,000 lb. load per tooth are in everyday use but their success required considerable development of the art of tooth design. In an article to appear next month, Mr. C. G. Pfeffer describes this development, presents some of the designer's pitfalls and shows how to get out of them.

PRODUCT ENGINEERING

GEORGE F. NORDENHOLT, *Editor*

The War's Impact On Design

WITH THE POWERS OF SCIENCE, engineering and industry of the United Nations pitted against those of the Axis partners, radical and sweeping changes in engineering design are taking place. As in all other countries affected by the present conflict, in the United States engineers are ably contending with the problems created by the scarcity of materials while striving to develop designs that excel in performance. Already the results of their ingenuity and efforts are startling and numerous. Almost daily some new development passes from the engineering department to the test department, on its way to the production line.

Practically all of the latest developments in engineering design relate to war equipment, therefore, not much concerning them can be published or even talked about. Most certainly this is not the time to give details of the design of the combat trainer made of steel and resin-bonded wood, or of any of the numerous other achievements of the design engineers. The fact that such developments are known to have been accomplished is prima facie evidence that startling new design techniques have been evolved and that we are entering a new era of engineering design.

But the greatest significance of the sweeping changes that are taking place in engineering design lies not in the changes themselves, but in the new points of view, neutral attitudes and methods of engineering design operations that are being firmly established. The new approach in design was neatly described by the chief engineer of one of the leading aircraft manufacturers when he said: "We were too complacent. We used to decide on a material that appealed to us because of one or two outstanding properties that it possessed. Having selected this material, we studied it, tested it and learned about it until we knew more about it than we did about any other material. Because the material gave satisfactory results, we stuck to it. Now, however, we are being forced to resort to other materials and, therefore, being compelled to learn to know them. After we know them intimately, we find that we can use them effectively and efficiently. Thus we have been obliged to take an entirely new attitude toward materials. We must learn how to use a great variety of them and not confine ourselves to a few pet materials. We are learning that best results are often obtained by using different materials in combination."

Thus the circumstances of war have aroused the design engineers to a fuller realization of the necessity of closely investigating the possibilities of many materials. This is perhaps not a fair statement because most engineers have long recognized that if they had sufficient data and a more complete knowledge of another material, they would be able to design for it effectively and efficiently. It would be more accurate to state that the present situation has given the design engineers an opportunity to show that if they are allotted by management the needed and reasonable appropriations for experimenting and testing to find out how to use another material, the results will be profitable. Design engineers are making full use of this opportunity and management is being convinced. Cost figures and performance cannot be disputed. Convictions that are based on such data are likely to be enduring.

SLEEVE BEARINGS – I

Factors That Govern Their Design and Performance

BRUNO SACHS
Technical Director, Johnson Bronze Company

MANY interdependent factors are involved in the design of bearings. The dependence of one factor upon another is sometimes difficult to determine, yet their proper coordination must be understood by an engineer before he can design with assurance a bearing that will perform satisfactorily under a given set of operating conditions. Primary considerations in bearing performance are heat equilibrium and bearing temperature.

All the mechanical work required to overcome friction in sleeve bearings is transformed into heat which raises the temperature of the bearing and its housing above the temperature of the surrounding atmosphere. The temperature rise at the bearing continues until stationary conditions of heat equilibrium are reached, that is, until the frictional heat produced per unit of time equals the amount of heat which is carried away from the bearing during the same time. This heat equilibrium, therefore, determines the stationary bearing temperature. The permissible limit of this temperature depends mainly upon the properties of the lubricating oil and those of the bearing material.

The heat quantity Q in B.t.u., or its mechanical equivalent W in ft. lb. which can be carried from a bearing, depends upon the volume, temperature and movement of the surrounding air. Calm outside air, of course, does not remove as much heat as does a strong ventilating draft such as is present around crosshead bearings and locomotive bearings. High values of heat dissipation can be obtained by water cooling the bearing by jacketing or spraying, and by forcing abundant quantities of oil through the running clearance.

Average values of W are shown in Fig. 1, for bearings without artificial liquid cooling, for a range of differences between the bearing and room temperature.

With p = unit bearing pressure in lb. per sq. in. of projected area
v = peripheral speed in ft. per min.
f = coefficient of friction.
d = dia. of shaft in in.
n = revolutions per min.

the mechanical equivalent W of the frictional heat in ft. lb. per sq. in. of projected bearing area produced per min. can be expressed as

$$W = pvf \tag{1}$$

$$W = \frac{p \pi d n f}{12} \tag{2}$$

In bearings with changing load, a differentiation must be made between the occurring maximum pressure and the mean pressure, the latter being determined over the time period of one load cycle. For the thermodynamical calculation the mean pressure is authoritative. On the other hand, in calculations involving oil film and fatigue resistance the maximum pressure must be used.

For conditions of heat equilibrium Equation (2) can be rewritten as

$$p d n = \frac{12 W}{\pi f} \tag{2a}$$

or

$$f = \frac{12 W}{\pi pdn} \tag{2b}$$

From the point of view of heat dissipation, the maximum permissible coefficient of friction is determined by Equation (2a) or (2b), if the stationary working temperature of the bearing is known. Oil companies, as a rule, recommend an operating temperature of 140 deg. F. in order to obtain best technical and economical results. With higher oil temperatures the rate of oxidation and deterioration increases rapidly, especially in circulating lubrication systems.

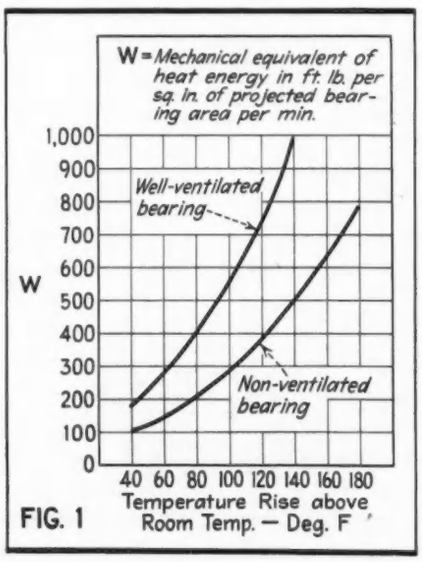


FIG. 1

Fig. 1—Average values of mechanical equivalent of heat energy dissipated by bearings operating at temperatures above room temperature

To simplify calculations Equation (2a) is plotted in Fig. 2 for a temperature rise, ΔT, equal to 80 deg. F., that is, a difference of 80 deg. F. between the bearing temperature and outside temperature, for both well ventilated and unventilated bearings. For example, these curves show that for p equal to 200 lb. per sq. in., d equal to 1 in., and n equal to 1,100 r.p.m., the maximum permissible coefficient of friction f is 0.007 for a well ventilated construction, and f' is 0.0035 for an unventilated bearing, if a bearing temperature rise of more than 80 deg. F. above the room temperature is to be prevented. For a bearing of 1¾ in. dia., having the same unit pressure and r.p.m., the maximum permissible coefficient of friction is 0.004 for a well ventilated bearing and is 0.002 for an unventilated bearing.

In Fig. 3 is shown the same relation between unit pressure p, r.p.m. n, dia. d and maximum permissible coefficient of friction as is shown in Fig. 2 but for a temperature rise of 140 deg. F. for well ventilated and non-ventilated bearings.

In the performance of a bearing, a decisive role is played by the coefficient of friction f. In sleeve bearings a differentiation must be made between dry friction of the shaft on the bearing, semi-fluid or mixed friction (often also called boundary friction), and fluid friction.

Dry friction takes place if no lubrication is provided or with complete failure of lubrication, and also at starting and stopping when the speed is zero. The value of the coefficient of dry friction varies from 0.16 to 0.29 depending upon the nature of the bearing and shaft materials, and their surface quality. Essential wear of the bearing or shaft takes place only with dry friction.

Semi-fluid or mixed friction takes place when the oil film is not able to bear the load. This state always exists if the quantity of the lubricant is small; such as is supplied by drop-feed cups, wick-feed oilers, bottle oilers, mechan-

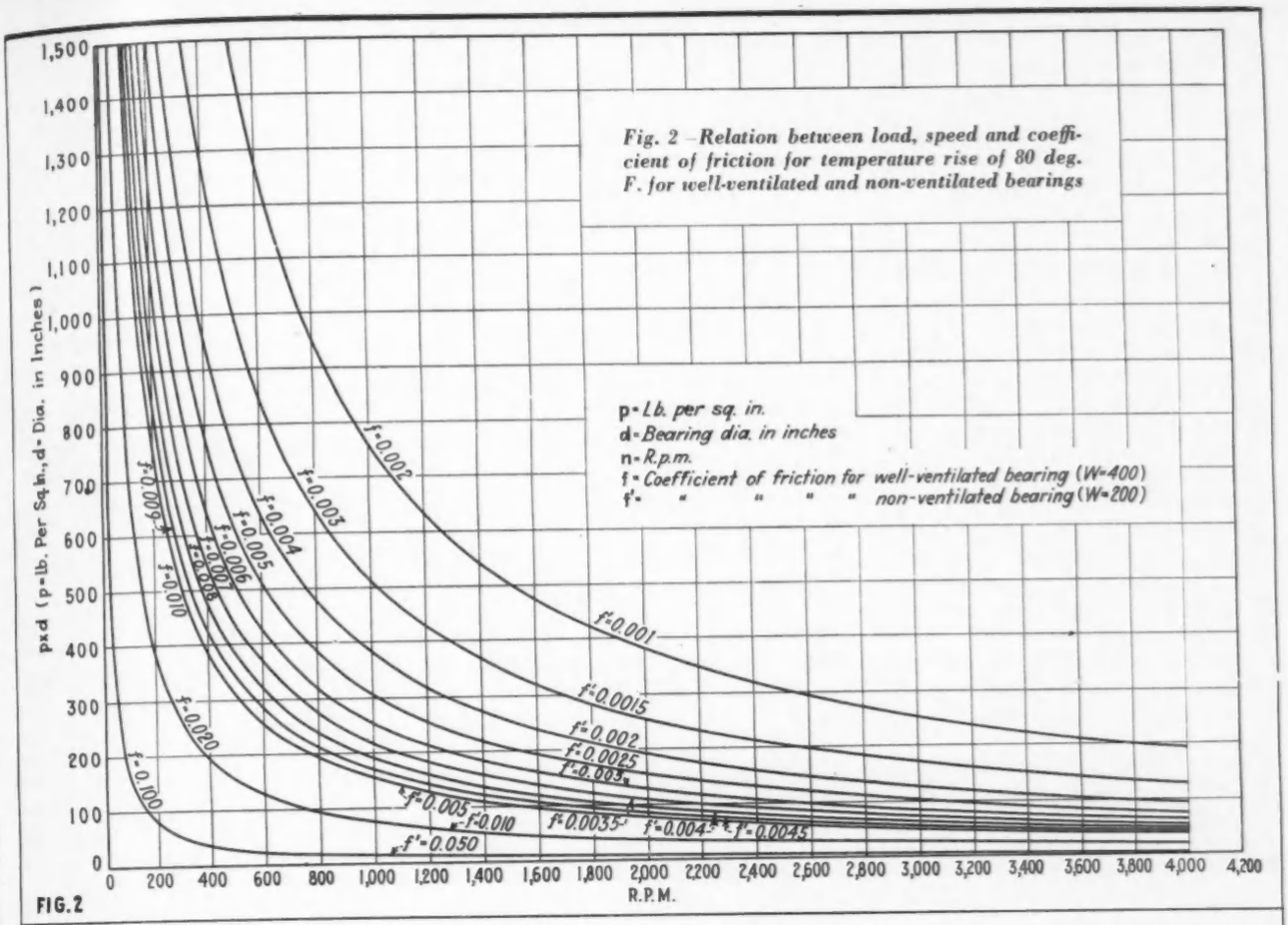


Fig. 2—Relation between load, speed and coefficient of friction for temperature rise of 80 deg. F. for well-ventilated and non-ventilated bearings
p = Lb. per sq. in.
d = Bearing dia. in inches
n = R.p.m.
f = Coefficient of friction for well-ventilated bearing (W=400)
f' = " " " " non-ventilated bearing (W=200)
pxd (p=Lb. Per Sq.In., d=Dia. in Inches)
R.P.M.
f=0.002
f=0.003
f=0.004
f=0.005
f=0.006
f=0.007
f=0.008
f=0.009
f=0.010
f=0.020
f=0.100
f'=0.001
f'=0.0015
f'=0.002
f'=0.0025
f'=0.003
f'=0.0035
f'=0.004
f'=0.0045
f'=0.005
f'=0.010
f'=0.050

FIG. 2

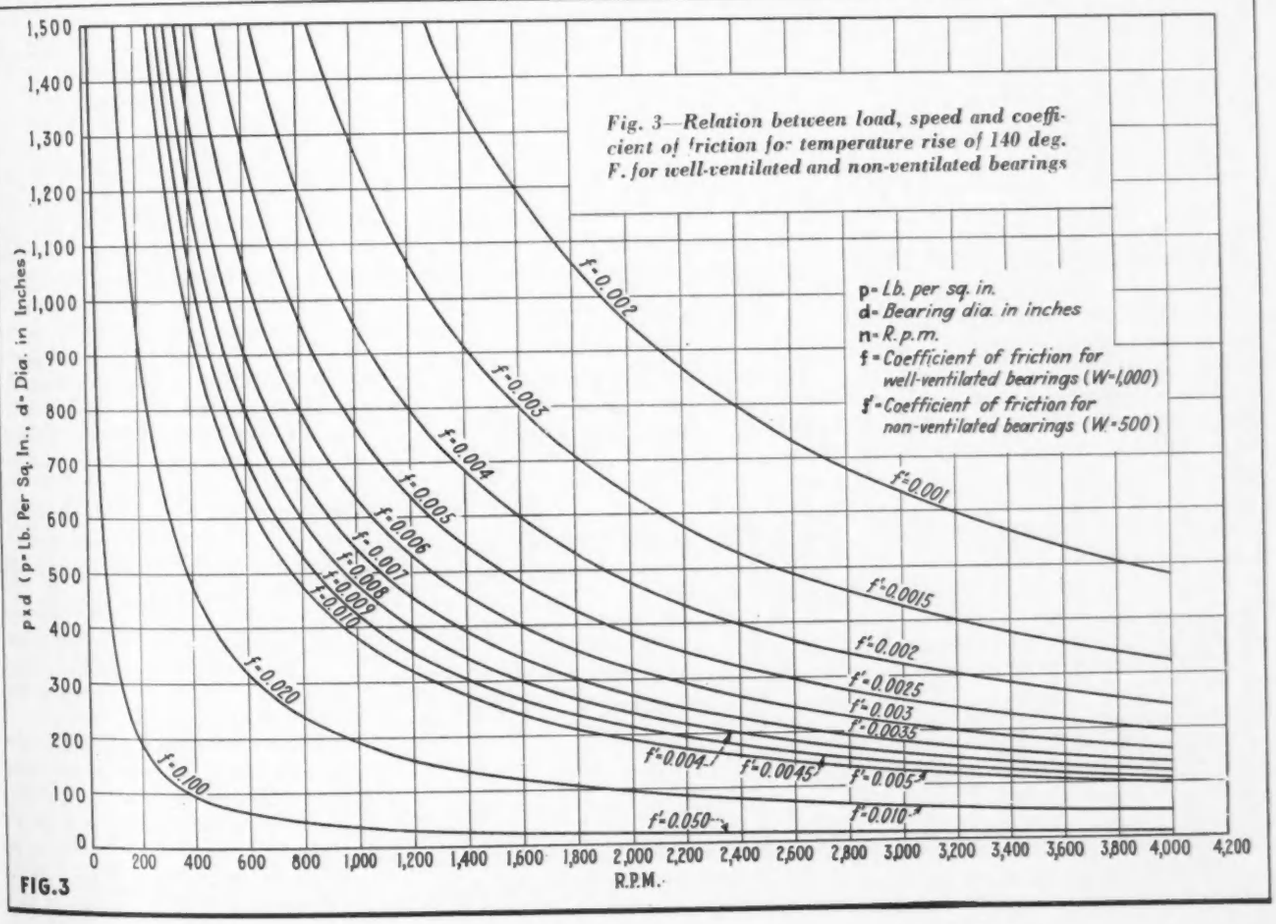


Fig. 3—Relation between load, speed and coefficient of friction for temperature rise of 140 deg. F. for well-ventilated and non-ventilated bearings
p = Lb. per sq. in.
d = Bearing dia. in inches
n = R.p.m.
f = Coefficient of friction for well-ventilated bearings (W=1,000)
f' = Coefficient of friction for non-ventilated bearings (W=500)
pxd (p=Lb. Per Sq. In., d=Dia. in Inches)
R.P.M.
f=0.002
f=0.003
f=0.004
f=0.005
f=0.006
f=0.007
f=0.008
f=0.009
f=0.010
f=0.020
f=0.100
f'=0.001
f'=0.0015
f'=0.002
f'=0.0025
f'=0.003
f'=0.0035
f'=0.004
f'=0.0045
f'=0.005
f'=0.010
f'=0.050

FIG. 3

ical force-feed lubricators and hand oiling. Semi-fluid friction also occurs in self-lubricating porous powder bearings.

Furthermore, this condition exists when the unit bearing pressure p is high and the speed low, or when the motion is oscillating or reciprocating like that found in journal joints and crosshead slides. A condition of semi-fluid friction, also, occurs on small areas of bearings running with fluid friction when there are local interruptions of the oil film caused by misalignment, vibrations, end leakage, bell mouthing, surface irregularities, inaccuracy of the shaft and bearing, entrapped foreign particles and incorrect oil grooves.

Even in bearings, properly designed, machined, installed, and fully lubricated, operating under periodically and quickly changing load, such as connecting rod bearings in high speed combustion engines, the limit of fluid friction might be reached or exceeded during short moments on minute surface zones when the load or its direction change jerkily, thus resulting in small semi-fluid states and in local rises of temperature far above the average bearing temperature, although the bearing as a whole runs in the state of fluid friction. Semi-fluid friction takes place also in bearings running with fluid friction during the stopping and starting periods. The value of the coefficient of semi-fluid friction varies between 0.02 and 0.12 depending upon operating conditions. Graphited and grease lubricated bearings usually run in a friction state between boundary and dry friction with a value between 0.12 and 0.16 for the coefficient of friction.

Most important for ideal mechanical performance is the state of fluid friction. This condition assures lowest friction values and practically does away with wear.

Some types of lubrication which deliver sufficient oil to the bearing to make fluid friction possible are pressure pump lubrication, splash oiling, bath oiling, ring, chain and collar oiling. In all these systems the oil circulates from a reservoir to the bearings and returns. The quantity of oil that flows through the bearing and whether or not it passes through an oil cooler, determines how much cooling of the bearing is accomplished by the oil.

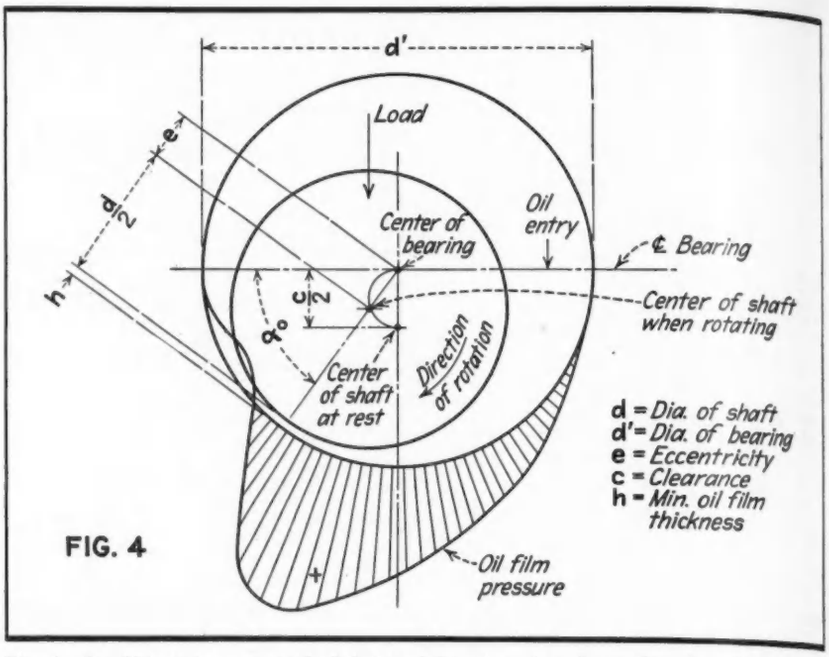


FIG. 4

Fig. 4—Conditions in a properly lubricated bearing after the wedge shaped load carrying oil film has been formed. Note how the shaft lifts out of contact bearing

Fluid friction is realized when the sufficiently supplied quantities of oil in the bearing are subjected to a hydrodynamical pressure necessary to maintain the shaft in a floating position; whereby a cohering film of oil is created which separates the shaft and bearing surfaces. The necessary pressure is obtained by designing and operating the bearing as a viscosity pump the action of which is based on the application of gap narrowing, that is, the formation of a wedge-shaped oil film. The value of the coefficient of fluid friction varies between 0.002 and 0.010 depending upon several factors.

Co-acting surfaces of bearings and shafts are not perfectly smooth, they have microscopic irregularities, valleys and peaks, and in order to prevent any contact of the two surfaces, the space between them must be filled at least with a film of lubricant having a thickness greater than is needed to prevent the surface peaks from touching.

One side of the film adheres to the bearing surface and is, therefore, stationary; the other side of the film adheres to the shaft and has a velocity equal to the surface speed of the rotating shaft; in other words, the velocity throughout the oil film layers increases from zero to the shaft speed. Hence, the friction is determined only by the internal resistance of the oil film layers. This resistance is proportional to the oil viscosity and the rate of layer speed change. The rate of layer speed change is expressed by dv/dh where dv stands for the layer speed differential and dh for the perpendicular distance between two adjacent layers.

Before continuing this analysis further a few words on viscosity may be timely. The unit of "absolute viscosity" in the c.g.s. system is the poise. A smaller unit is the centipoise which is equal to one hundredth of a poise. Numerical values of the "absolute viscosity" in centipoises are obtained by multiplying the "kinematic viscosity," expressed in centistokes, by the oil density in grams per cu. cm. at the temperature in question.

A commercially used method, known as Saybolt Universal Seconds for measuring and describing the oil viscosity, is the time in seconds it takes an oil at a given temperature to flow through a standard size orifice and fill a standard size cup.

Table I—Viscosity Range of S.A.E. Numbered Oils

S.A.E. Viscosity Number	Viscosity Range Saybolt Universal Seconds			
	At 130 deg. F.		At 210 deg. F.	
	Minimum	Maximum	Minimum	Maximum
10	90	120		
20	120	185		
30	185	255		
40	255			75
50			75	105
60			105	125
70			125	150

Saybolt Universal Seconds $S\ U$ can be transformed into centistokes thus

When $S\ U \leqq 100$,
centistokes $= 0.226\ S\ U - 195/S\ U$
When $S\ U \geqq 100$,
centistokes $= 0.220\ S\ U - 135/S\ U$

For bearing calculations the following approximate values of the specific gravity or density of oils at 60 deg. F. can be used

0.88 for S.A.E. 10 to 30
0.89 for S.A.E. 40 to 60
0.90 for S.A.E. 70

The specific gravity of oils changes with the temperature according to:

$$\text{Specific gravity at } t \text{ deg. F.} = \frac{\text{specific gravity at 60 deg. F.}}{1 + 0.00036\ (t - 60)}$$

Diagrams showing for each grade of oil the relation between viscosity and temperature can be obtained from oil companies. These charts (A.S.T.M. Standard D-341-39) are so constructed that for any given petroleum oil the viscosity-temperature points lie on a straight line within that temperature range in which the oil is a liquid.

The often referred to S.A.E. Viscosity Numbers according to their definition do not describe in any way the rate of change of oil viscosity with the temperature but, as shown in Table I, merely establish the range of oil viscosity at one arbitrarily chosen temperature.

When stationary the shaft, of course, rests on the bottom of the bearing. When rotated, however, the shaft when properly lubricated lifts out of contact with the bearing and is displaced in the direction of rotation as shown in Fig. 4, and rides on a film of lubricant. The formation of the wedge shaped load carrying oil film takes place under well defined conditions. It is the designer's task to secure these conditions.

With reference to Fig. 4 let

c = clearance in in.
h = minimum oil film thickness in in.
d' = dia. of bearing in in.
d = dia. of shaft in in.
l = length of bearing in in.
e = eccentricity in in.
α = complement to the angle of radial displacement of the rotating shaft

Then arbitrarily let

f = coefficient of fluid friction
p = unit bearing pressure in lb. per sq. in. of projected area
n = revolutions per min.
Z = "absolute viscosity" in centipoises of the oil at its actual working temperature in the bearing.

C_1, C_2, m = constants from Fig. 5 which are functions of α. To each value of C_1, belongs one certain value of C_2 and of m.
G = Constant from Fig. 6 which depends on the ratio l/d.

Then these equations are valid

$$c = d' - d \tag{3}$$
$$h = (c/2) - e \tag{4}$$
$$e = (c/2)\ C_1 \tag{5}$$
$$c = \left(\sqrt{\frac{C_2}{G}}\right)(1.233)\ (10^{-4})\left(\sqrt{\frac{(Zn)}{p}}\right)d \tag{6}$$
$$h = \left(\frac{1 - C_1}{2}\right)\left(\sqrt{\frac{C_2}{G}}\right)(1.233)\ (10^{-4})\left(\sqrt{\frac{Zn}{p}}\right)d \tag{7}$$
$$f = m\left(\sqrt{\frac{4d}{l} + 1}\right)(1.233)\ (10^{-4})\left(\sqrt{\frac{Zn}{p}}\right) \tag{8}$$

The analysis and interpretations of Equations (6), (7) and (8) and their interdependence is best done graphically. For this purpose they are transformed by dividing Equation (7) by Equations (6) and (8) respectively, taking the form

$$\frac{h}{d} = \left(\frac{1 - C_1}{2}\right)\left(\frac{c}{d}\right) \tag{6a}$$
$$= \left[\frac{(1 - C_1)\ \sqrt{C_2}}{2\ \sqrt{G}}\right](1.233)\ (10^{-4})\left(\sqrt{\frac{Zn}{p}}\right) \tag{7a}$$
$$= \frac{(1 - C_1)\ (\sqrt{C_2})\ f}{(2\ m\ \sqrt{G})\left(\sqrt{\frac{4\ d}{l} + 1}\right)} \tag{8a}$$

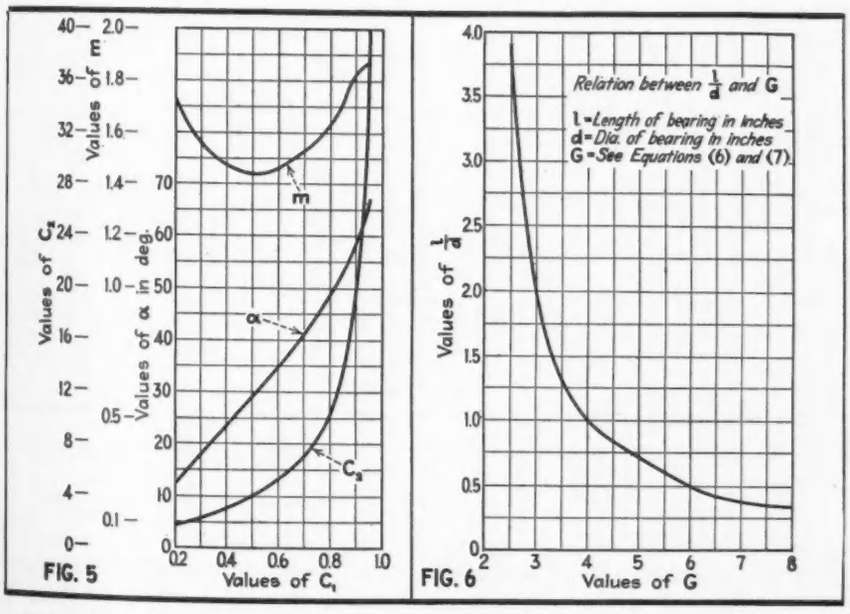
For a bearing with l/d equal to 1, the Equations result in

$$\frac{h}{d} = \left(\frac{1 - C_1}{2}\right)\left(\frac{c}{d}\right) \tag{6b}$$
$$= (1 - C_1)\ (\sqrt{C_2})\ (0.30825)\ (10^{-4})\ \sqrt{\frac{(Zn)}{p}} \tag{7b}$$
$$= \frac{(1 - C_1\ (\sqrt{C_2})\ f}{8.9444\ m} \tag{8b}$$

In Fig. 7 the Equations (6b), (7b) and (8b) are shown graphically with the constants C_1 as abscissa and h/d as ordinates. The scale for these ordinates is so chosen that the dimensions of the oil film thickness are given in micro-inches, (1 micro-inch equals 0.000001 in. or 1 times 10^{-6} in.), the clearance and diameter given in inches.

Each equation is represented by a group of curves and the relations between c/d, h/d, $\sqrt{Zn/p}$ and f for bearings having a ratio l/d equal to 1 can be readily read off. For instance, if c/d equals 0.0015 in. and $\sqrt{Zn/p}$ equals 16, the coefficient of friction f will equal 0.007 and h/d equals 535. Assuming, as an example, that d equals 1 in., the oil film thickness will be 535 micro-inches. If now the value of $\sqrt{Zn/p}$ becomes smaller, for instance during the stopping period, the corresponding values for f and h are to be found by moving down along the line c/d equals 0.0015 in. Now for $\sqrt{Zn/p}$ equal to 7, it can be seen that f becomes 0.0032 and h equals 130 micro-inches. If the sum of the maximum heights of the surface irregularities of shaft and bearing is of the order of 130 micro-inches, mixed or semi-fluid friction would start at this point.

The significance of the size of the clearance can be easily recognized from Fig. 7; for example, with a bearing having d equal to 1 in., a clearance of 0.0005 to 0.0010 in. and values of $\sqrt{Zn/p}$ of 6.3 to 7.5, the coefficient of friction f is 0.003 with an oil film thick-



Figs. 5 and 6—Values of constants used in determining the thickness of oil film

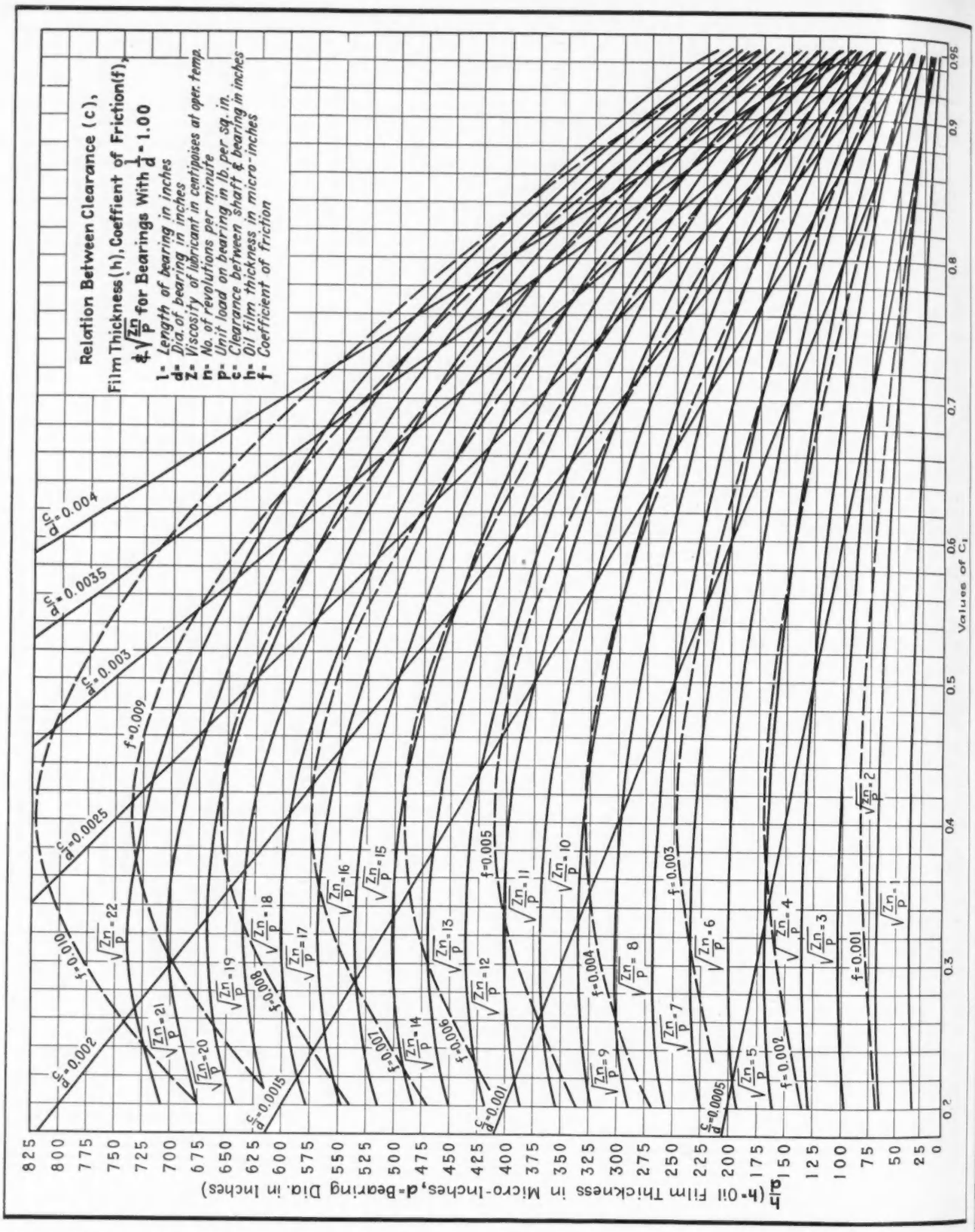


Fig. 7—Clearance, film thickness, coefficient of friction and other factors for bearings having a length diameter ratio equal to one

ness h equal to 205 to 225 micro-inches. With a clearance of 0.0025 to 0.0030 in. it will be practically impossible to obtain this coefficient of friction of 0.003, because the oil film thickness will become so small that surface contact between shaft and bearing will throw the bearing from the fluid friction state into mixed friction.

Furthermore, it can also be seen from the curves in Fig. 7 that with an enlarged clearance in the bearing as a consequence of wear, the oil film thickness becomes smaller and the safety against surface touching decreases.

BRIGHTNESS RATING

Of Metal Surfaces Measured by Reflected Light

ALFRED M. SUGGS

P. R. Mallory & Company, Inc.

MANY PROBLEMS encountered in the metallurgical field need for their solution the determination of the light absorption of a metal surface and the rate at which a tarnish of film appears on the surface. There are numerous instances in which such information is extremely valuable. In electrical circuits, wherein the contacts are open for appreciable periods of time the contacts should be of materials which will not tarnish rapidly, as a heavy tarnish film usually results in high resistance contacts. Also, such a film often mars the appearance of the instrument. In applications such as ornamental ware and table ware, the appearance is of prime importance in the choice of material.

Several methods have been used to measure the specular and diffuse reflection from metal surfaces, and these measurements have been used in various ways to give the surface under consideration a brightness rating. However, these methods require considerable equipment and time for set-up and may require several calculations to arrive at a brightness rating.

The brightness rating, as determined by the instrument described herein, is direct reading and is proportional to specularly reflected light and to diffusely reflected light which is deviated only a small amount from the angle of reflection. The design of the instrument is based on the assumption that the light scattered through a small angle from the angle of reflection will make the surface appear brighter, while the light scattered through wide angles does not add materially to the brightness of the surface.

A schematic diagram of the instrument is shown in Fig. I. At *A* is a concentrated light source which consists of a double-filament automobile headlamp in which both filaments are heated. The filaments are operated slightly under their rated current to insure a long life of the bulb and a more constant light source. The surface to be measured is held in place at *B* by a clamp and locating fixtures so that the surface can be accurately registered for subsequent measurements. Light reflected from the surface strikes the photronic cell *C* after passing through the ground glass plate at *D* which transmits approximately 75 percent of the incident light. The object of this plate is to diffuse the light so that the location of the sample will be less critical. The plate also serves to pick up light scattered by the sample and redirect it so that a portion of the light diffused by the sample reaches the photronic cell. At *E* is shown the filter holder which will hold two standard light filters. With proper selection of filters, the spectrum can be divided into its various colors, and the absorption of the tarnish film for each color can be measured. Compartment *F* contains the current supply and controls for light source. Current generated by the photronic cell is measured on a 0–50 microammeter which is not built in with the rest of the instrument.

Since the results obtained are comparative and not absolute, it is necessary to use some surface as a standard to which others can be compared. A silver-back glass mirror serves satisfactorily for this purpose. If it is desirable to compare one metal with another, fine silver with a finish similar to the test sample can be used as a standard substance.

Several sets of measurements in which the samples were exposed to the atmosphere have been completed. The results so far indicate that this instrument is much more sensitive than the eye to changes on a metallic surface. In measurements made with the filters, the results usually agree with what would be expected from a visual inspection. For example, the tarnish film on one sample had a distinct yellow cast and the absorption was found to be 8 percent greater for white light than for yellow light.

Curves, Fig. 2, obtained by use of this instrument, show changes in brightness resulting from exposure to air at room temperature of highly polished surfaces and dull surfaces on fine and coin silver samples.

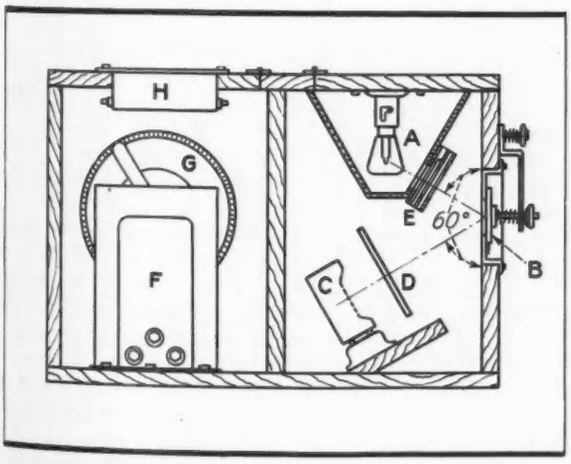


Fig. 1—Schematic diagram of instrument for determining tarnish rate of metal surfaces

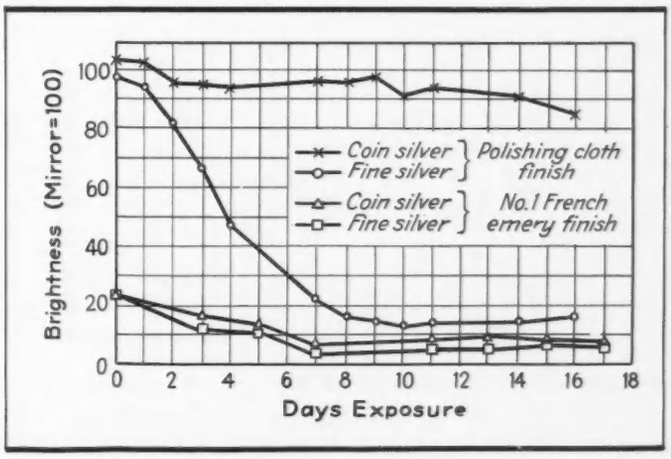
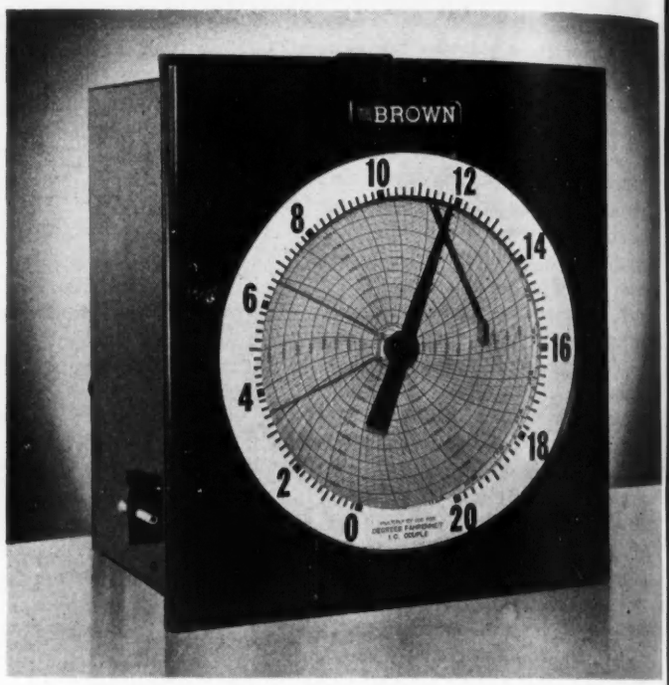


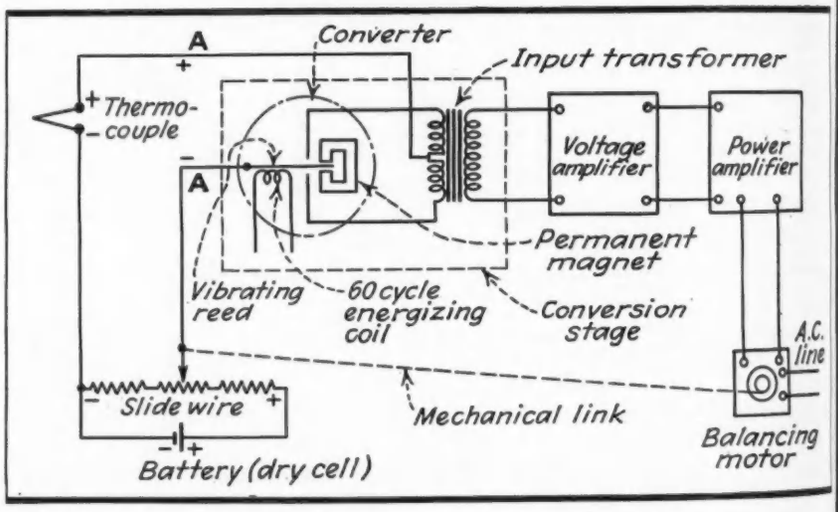
Fig. 2 –Tarnish rate of fine and coin silver samples, showing effects of two different surface finishings

Modern Designs

Electronic Unit Eliminates Cyclic Mechanism

Galvonometers—for years the "brains" of self-balancing potentiometers — have been replaced by a compact electronic unit in this new continuous balance potentiometer pyrometer, designed by Brown Instrument Company as the forerunner of a series of industrial instruments which will use the electronic balancing system. In the unit, unbalanced voltage of the measuring circuit is converted to alternating voltage, amplified, and used to drive a motor in the proper direction to balance the circuit and correct the instrument reading. Results: Instrument corrects itself instantly and continuously regardless of vibration or position. When temperature is stable there is no mechanical motion. Galvanometer inertia, pivot friction and complicated cyclic cam and lever mechanism are gone. Principle design problems were selection and design of electrical and mechanical components for simplicity, long accurate life.





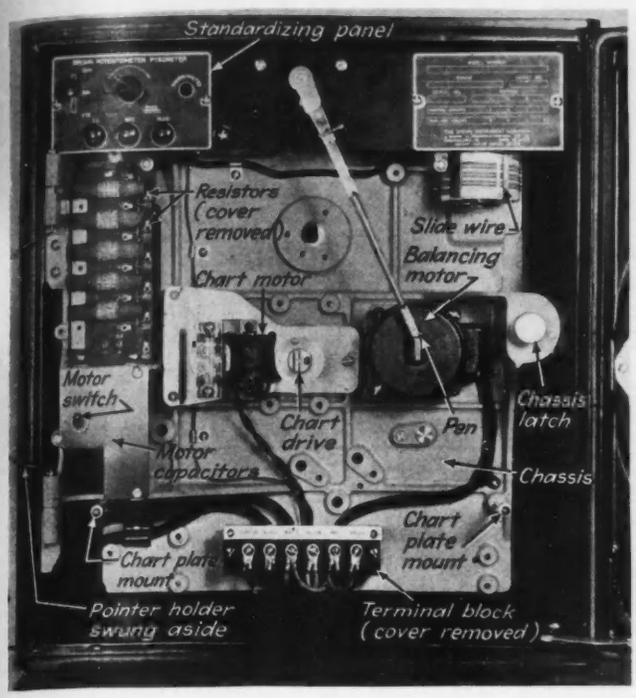
Measuring circuit of balancing system consists of thermocouple and slide wire through which known current from dry cell is passed. When thermocouple voltage changes, an unbalanced voltage appears across *AA*. Conversion stage of the system changes this d.c. unbalanced voltage into a.c. which then feeds into amplifier circuit.

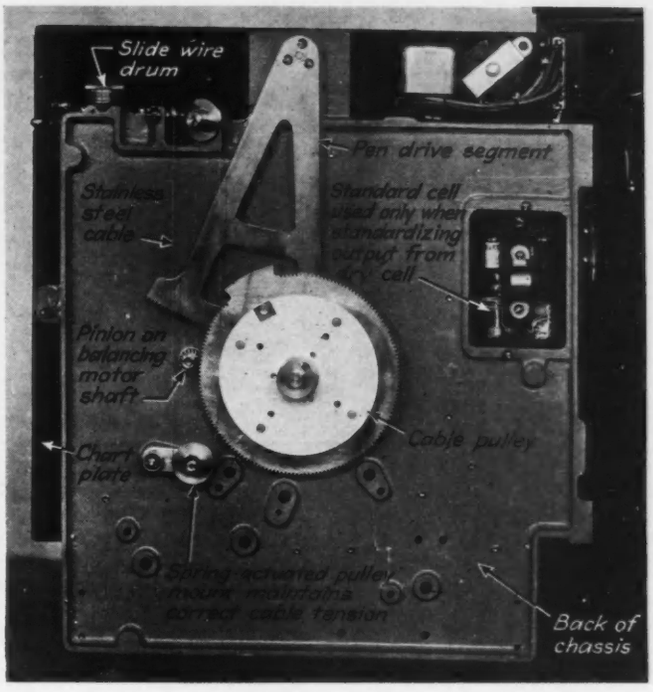
Converter is single pole, double throw switch actuated by a.c. energizing coil so that it vibrates in synchronism with line voltage. Contacts mounted on vibrating reed throw unbalanced voltage into alternate ends of transformer primary. This 60-cycle control voltage is either in phase or 180 deg. out of phase with line a.c., depending on polarity of unbalanced voltage from measuring circuit.

Voltage amplifier is conventional 3-stage resistance-capacitance coupled circuit. It supplies greatly amplified voltage of same wave shape as control voltage to grids of power tubes. Power amplifier is special circuit.

Balancing motor is reversible, two phase induction type. One winding is energized continuously by line voltage; the other is energized by the power amplifier. When thermocouple circuit is unbalanced, phase relationship between two motor windings is determined by direction of unbalance, hence direction of motor is such as to correct slide wire resistance to balance the circuit, at the same time correcting instrument reading and operating controls.

When thermocouple circuit is balanced, its voltage is zero and power amplifiers deliver only pulsating d.c. to balancing winding of the motor. This brakes the motor at true balance and prevents overtravel. Mechanical motion occurs only when correction is necessary. Maximum driving power is obtained with unbalance of less than 0.06 percent of scale span, or equivalent to 0.000005 volt unbalance in the thermocouple circuit.

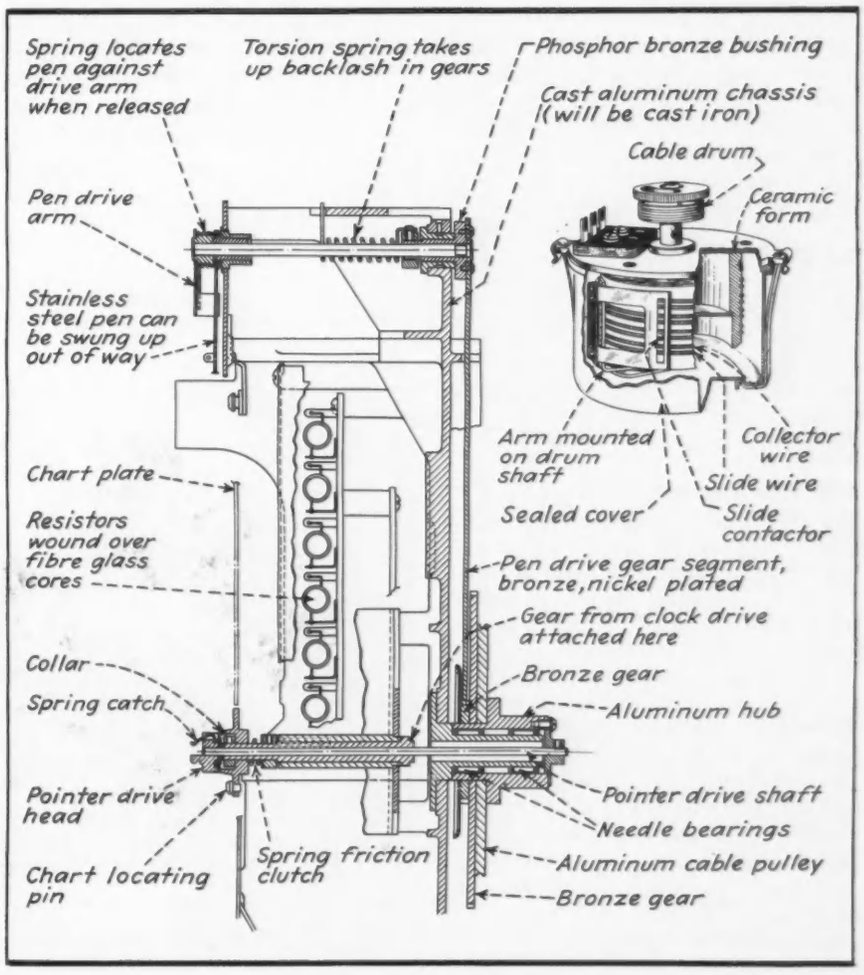



Slide wire drum
Pen drive segment
Stainless steel cable
Standard cell used only when standardizing output from A.C. cell
Pinion on balancing motor shaft
Cable pulley
Chart plate
Spring-actuated pulley mount maintains correct cable tension
Back of chassis

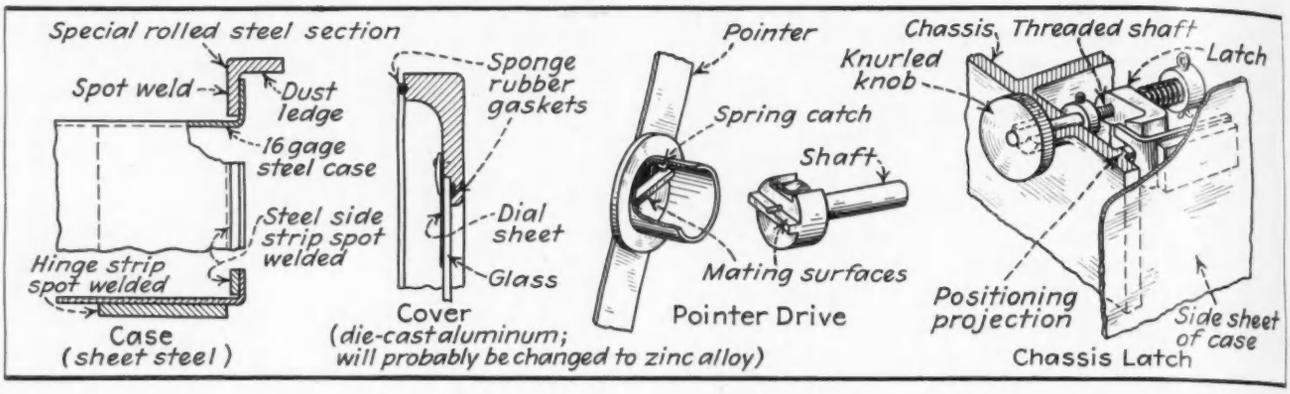
Elements of instrument are designed as compact, interchangeable units mounted for easy access. Balancing motor, chart motor, slide wire, terminal block, resistors and motor capacitors are mounted on front of chassis, which swings out to give access to units mounted on back. Continuous balance electronic unit and dry cell are mounted inside back of case (not shown). Covers that are frequently taken off have screw slots so that screws need not be completely removed. At left is shown front of chassis; at right chassis is swung out to show back.

Drive mechanism moves slide wire contactor, pen and pointer in unison and is simpler than levers and cams formerly required with galvanometer type instruments. Balancing motor drives hub mounted in needle bearings on back of chassis. Pointer shaft extends frontward from this hub through center of chart drive shaft which is rotated by chart motor. Pen shaft turns in phosphor bronze bushing.

Slide wire is driven by 9-strand stainless steel cables wrapped around pulley on hub. Cable passes over idler pulleys and is wrapped 3½ times around drum on slide wire unit. Ends are secured to drum. Slide wire, 25 in. long and with 1,700 convolutions, is bare Manganin wire, space-wound over Formex insulated wire core. Close control over production gives accurate linear resistance of the wire. Resistance of complete unit is standardized by adjusting number of turns in shunt-connected resistance wound in groove at top of ceramic form. Collector is bare nickel-copper wire space-wound over copper core and is helically wound alongside slide wire. Silver contactor rides between slide wire and collector wire and is driven around helical windings by slotted arm. Contactor has sliding action on wires, but rolls as it moves up and down in slot. This wiping action keeps contactor clean, prevents uneven wear. (*continued on next page*)



MODERN DESIGNS – Electronic Balancer *(continued)*



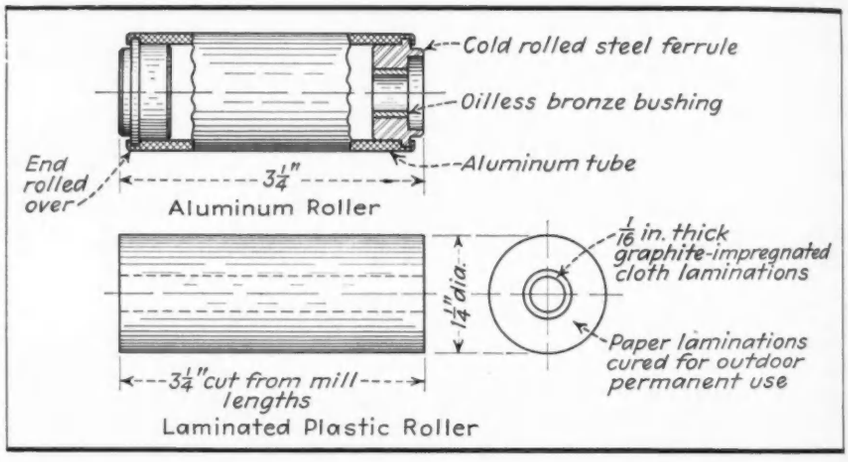
Steel case, fabricated by spot-welding, is dust and moisture proof, has sturdy reinforcing frame around front. Finish of case is black wrinkle enamel, chassis is light gray wrinkle finish.

Chassis latch is unique. Twist of knob turns dog to right until it strikes positioning pin behind lug which is spot-welded to case. Another twist pulls the chassis tight against the lug.

Pointer drive is designed to give accurate positioning free from back-lash and yet is designed so that the pointer can easily be released from the drive shaft and swung back out of the way.

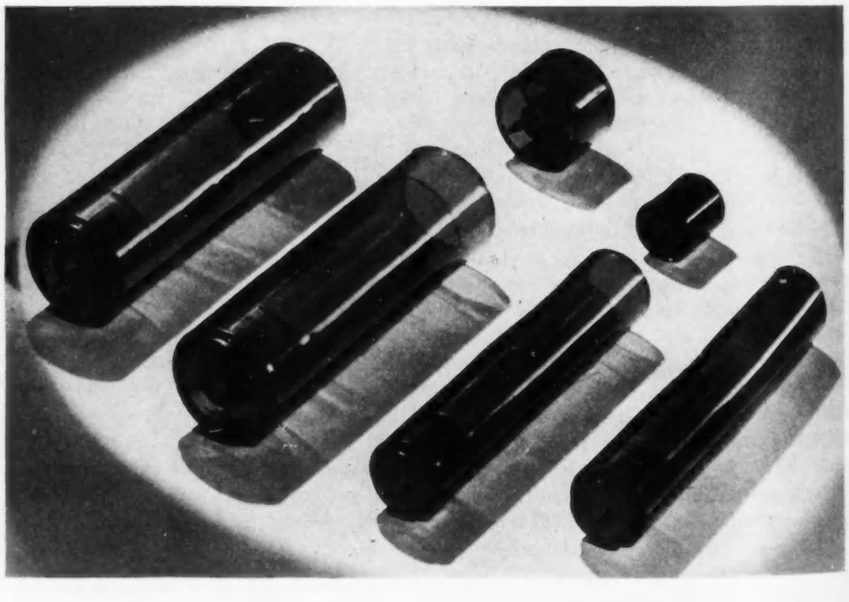
Designs of Plastic Rollers

Aluminum tube, two cold rolled steel ferrules, and two porous bronze bushings are replaced by one laminated phenolic plastic tube in the redesign of rollers for Gilbert & Barker gasoline pumps. In the pump, the ¾ in. synthetic rubber hose is looped around a weighted pulley inside the housing, and runs over seven rollers up to and at the point where the hose emerges from the housing. Roller sizes vary from 1½ to 6 in. long. Inner core (⅛ in. wall) is graphite-impregnated cloth laminations, over which are wound impregnated paper laminations. Tube is purchased in 36 in. lengths and is cut off to specifications. Plastic rollers will not stick to rubber even in hot weather, which was principal problem with metals other than aluminum. Design change saves 30 in. of aluminum tubing and 14 bronze bushings per pump.



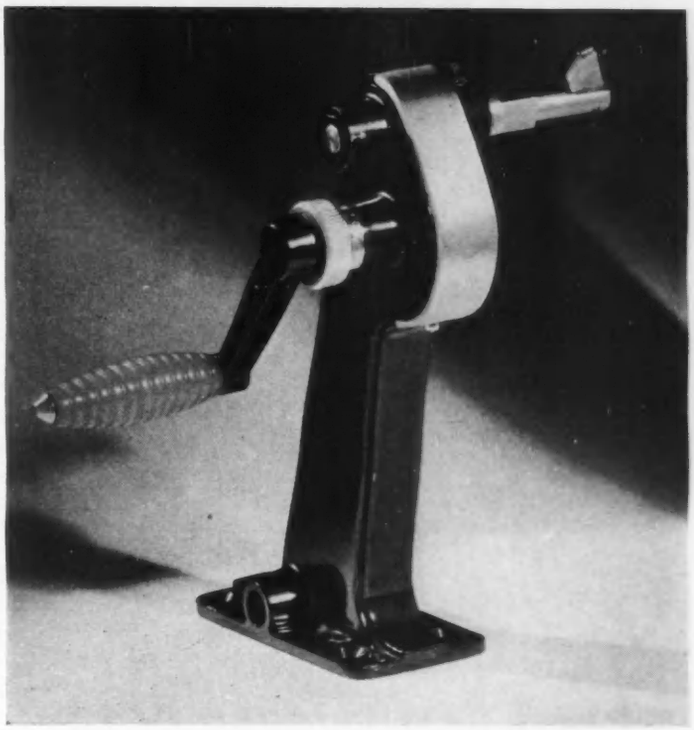
Machine gun ammunition feeds to the guns from the magazine over these plastic rollers in the world's fastest and most heavily armed dive bombers. Material is cellulose acetate butyrate (Tenite II), selected for its ability to withstand strain of constant wear and hard blows. Each roller consists of molded tube and molded bearings which are pressed into tube ends up against molded-in shoulders. Rollers are molded by Columbus Plastic Products, Inc., for Curtiss-Wright Corporation.

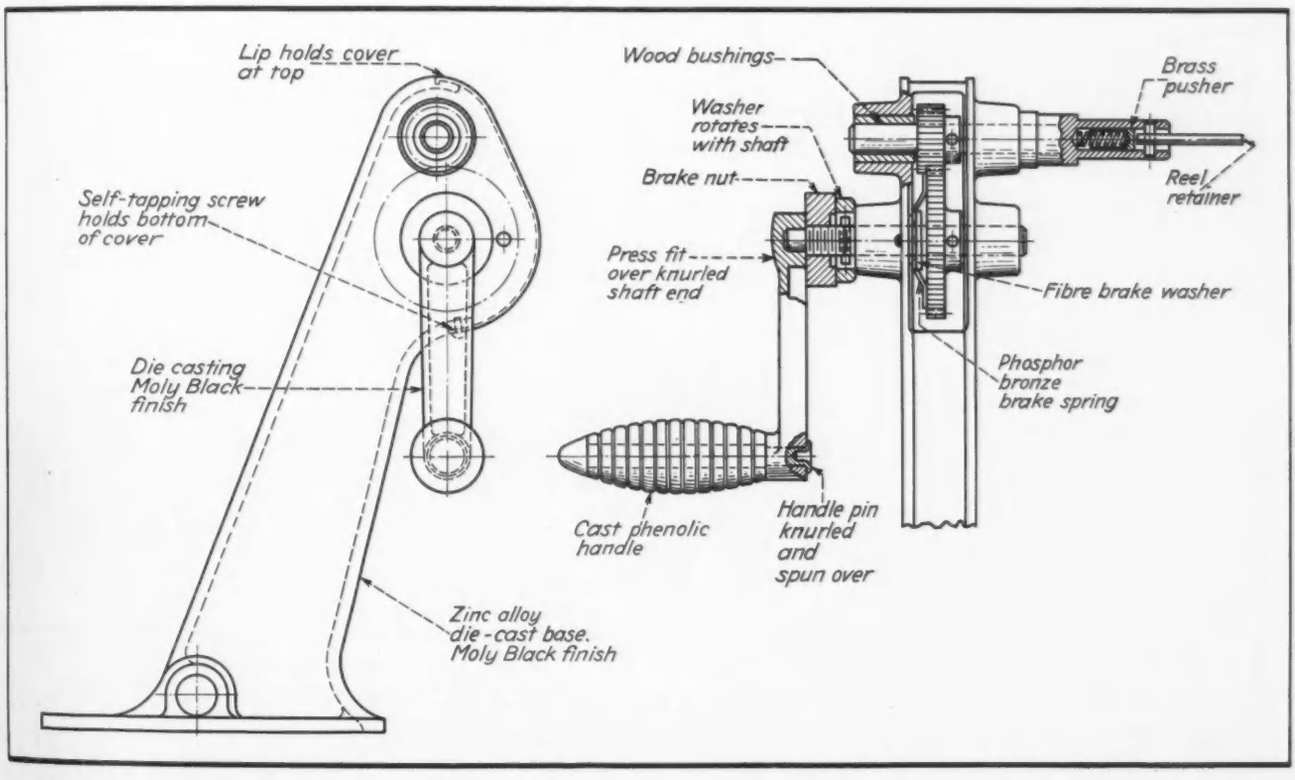
Other cellulose acetate and butyrate plastic parts in aircraft are control stick handles, instrument dials, dehydration tubes, electric light lenses, bomber visors and tank covers.



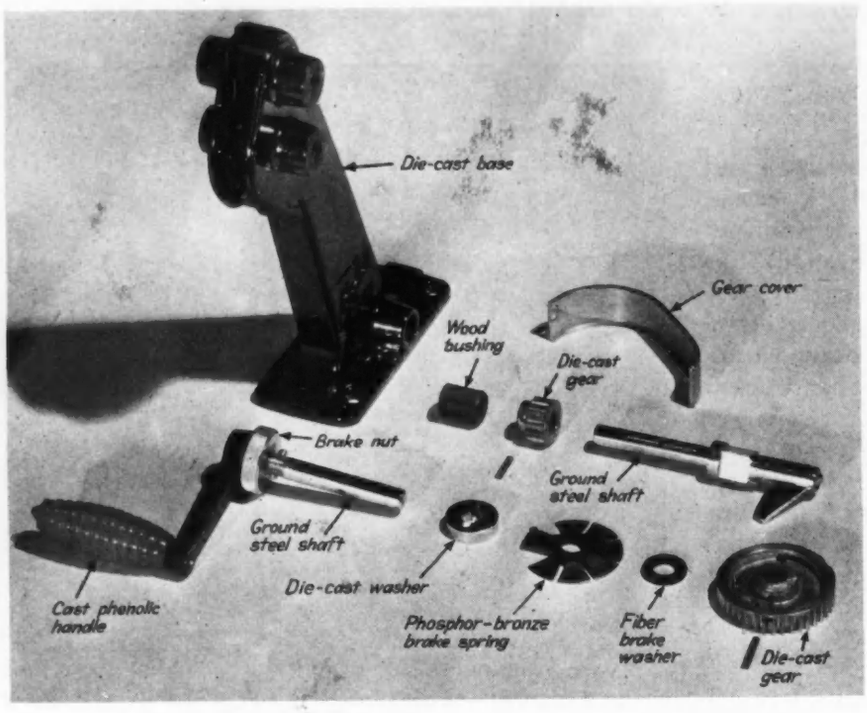
Two-Stage Brake Prevents Coasting of Film

Careful design of gears and bearings, and unique design of brake give this American Bolex motion picture film winder its principal feature—precise and smooth operation. The units are used in pairs for winding 8 or 16 mm. film from one reel to another to facilitate splicing and editing. All parts are designed for assembly into either right- or left-hand units. Adjustable brake is designed to give correct tension control so that film can be rewound as fast as desired with no "free-wheeling" over-travel; both hands can be freed and the film will stay tight. The unit was designed by Whitehouse Leather Products Company, who became interested in photographic equipment while making leather camera bellows.




Lip holds cover at top
Self-tapping screw holds bottom of cover
Die casting Moly Black finish
Zinc alloy die-cast base. Moly Black finish
Wood bushings
Washer rotates with shaft
Brake nut
Press fit over knurled shaft end
Brass pusher
Reel retainer
Fibre brake washer
Phosphor bronze brake spring
Cast phenolic handle
Handle pin knurled and spun over

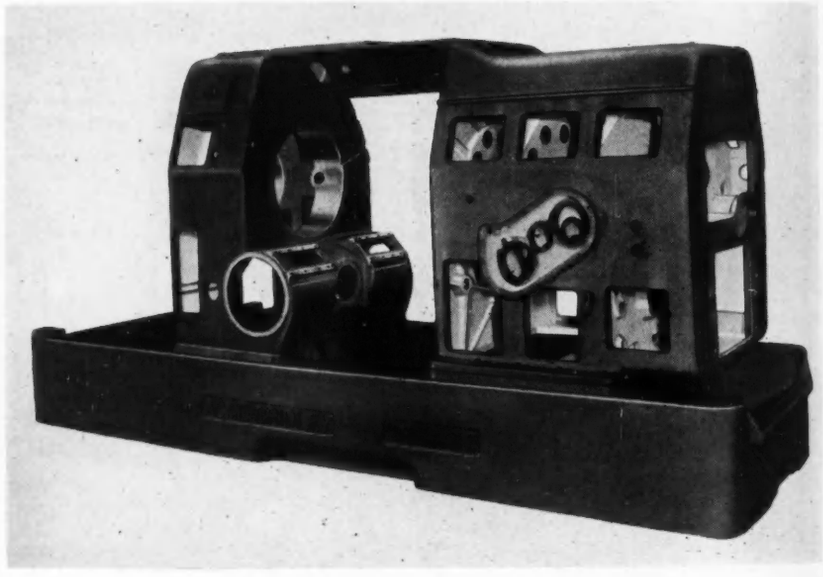
Tension control consists of two-stage brake. For light tension, knurled nut is slightly tightened, moving the shaft and large gear. Side of gear presses against brake spring, giving light braking action. As locking nut is tightened further, gear is forced against fibre washer, giving strong braking action. The washer between knurled nut and housing is rotated with shaft by pin, and prevents knurled nut from loosening as handle turns. Assembly methods are simple, thorough. Wood bushings are pressed into die-cast base and are reamed for accuracy. Gears are pinned to shafts by small grooved pins. Die-cast handle arm is pressed on knurled end of drive shaft, producing joint that is stronger than shank of arm. Handle pin is knurled, pressed into arm, and riveted over. Gear cover is fastened at one end by lip which fits into groove in die-casting; at other end by self-tapping screw.

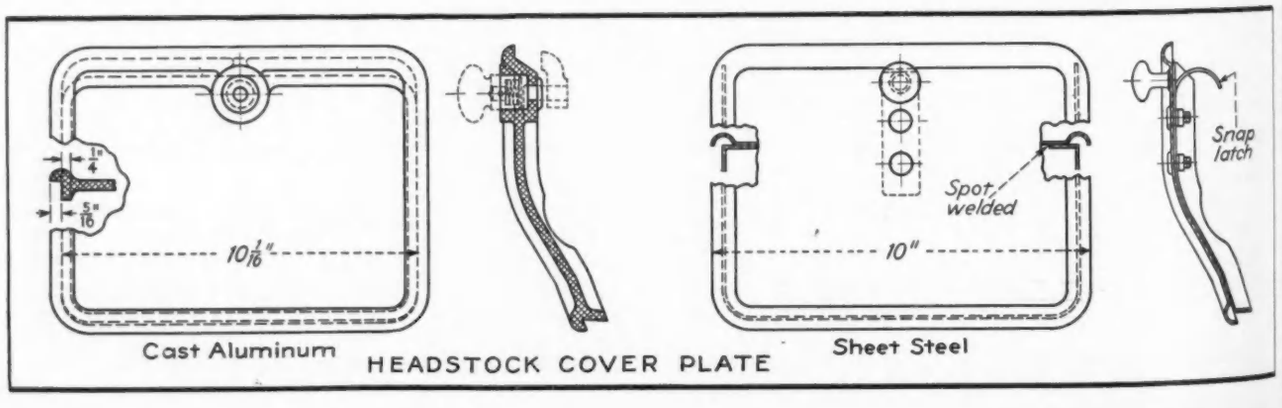


Base is die-cast zinc alloy, with all holes reamed for accuracy. Bearings are oilless wood bushings (Arguto) pressed into housing. Shafts are ground to tolerance of 0.0005 in. Gears are zinc alloy die-castings. Brake consists of phosphor bronze spring backed up by vulcanized fiber washer. Because of materials shortages, next production order will probably specify spring steel in place of phosphor bronze. Handle is attractive red cast phenolic plastic, shaped to keep fingers from slipping. Handle arm is zinc alloy die-casting, as is the thick washer mounted behind the knurled nut on the handle shaft. Finishes are Moly Black plating for the die-cast base and handle, and chromium plating for other metal parts. Gear cover is satin finished chromium plated. Tradename letters are raised on base. Appearance of unit is attractive, with black, chromium, and red in well-balanced contrast. The winders can be mounted in wood blocks to give added height necessary for large reel sizes.

Steel Covers Save 100 lb. Aluminum per Machine

Twenty cast aluminum cover plates on Acme-Gridley bar automatic machines made by National Acme Company have been changed to 18-ga. sheet steel fabricated by spot welding. Aluminum savings amount to over 100 lb. per machine and total around 15,000 lb. per month. While stamped sheet steel covers are not new, design of these covers, some of which are complicated shapes, is unique. Rounded covers add to appearance, and automatic welding operations have reduced costs. Fitting is simplified because steel covers can be hammered to shape whereas aluminum castings had to be machined or ground. Contributing to low cost is a snap latch in place of multiple part rotating catch which was used on aluminum covers. Machine base with cover removed is shown at right; typical cover is illustrated below.

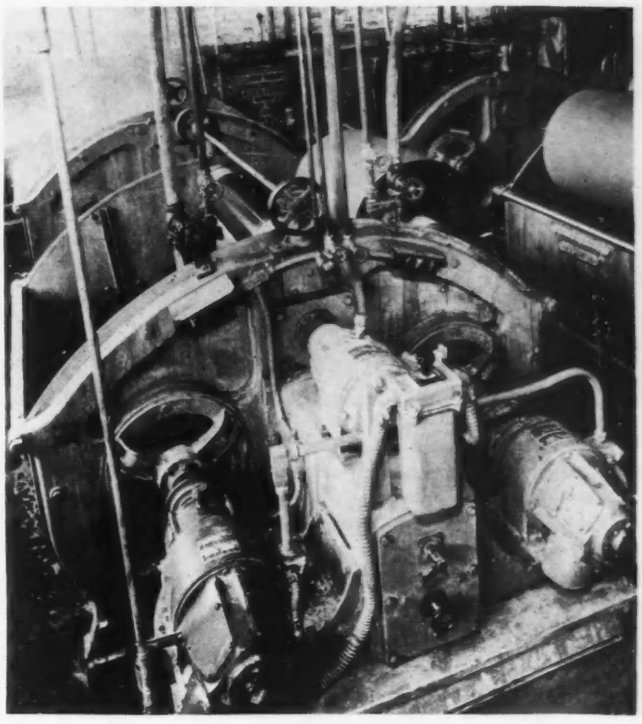


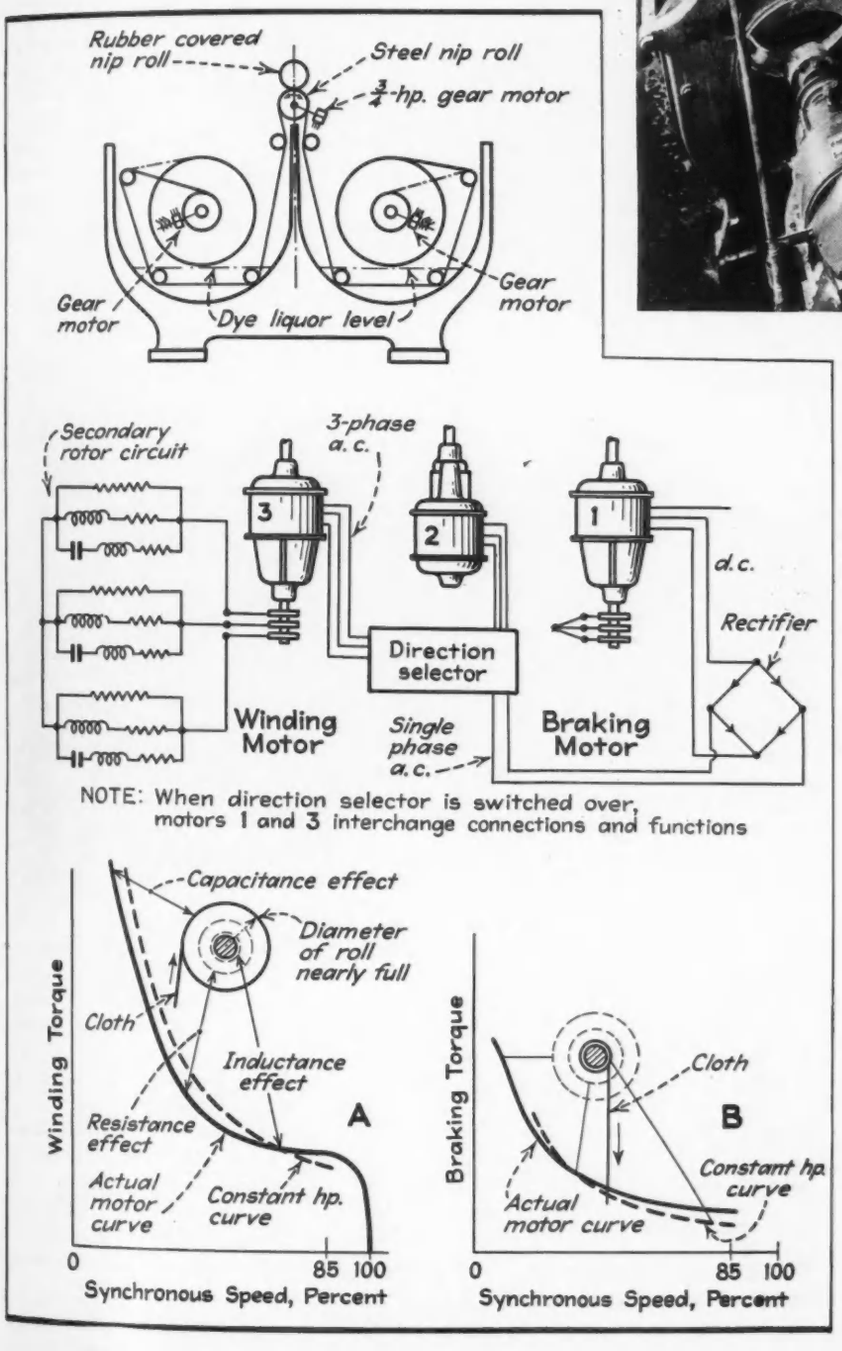


HEADSTOCK COVER PLATE

Motor Drive Gives Uniform Cloth Tension

Uniform dyeing of cloth requires uniform speed and constant tension, often difficult to maintain because of changing roll diameters. In the dye jig of Duo Dyeing Machine Company, where cloth receives dye treatments in two separate compartments while being run back and forth from one roll to another, constant tension is maintained by an ingenious General Electric three-motor drive in which constant horsepower characteristics are automatically regulated by special secondary circuits. Motors are splash-proof, fan-cooled, induction type connected to rolls through flexible couplings. Master reversing switch, tension selector switch, start and stop timing controls, and other elements of the electrical circuits are encased in sturdy drip-proof housings at motor end of jig.



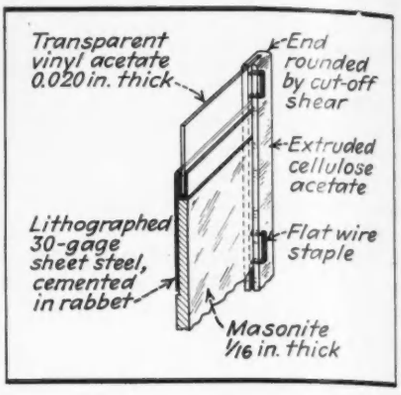


Constant speed of about 80 yd. per min. is maintained by ¾-hp. induction motor which drives nip rolls. Constant tension is provided by two other motors, one to drive the winding roll, the other to brake the unwinding roll. These motors are duplicate induction motors with wound rotors, and operate with constant horsepower characteristics so that braking and driving torques are proportional to roll diameters. Motor speeds vary from 85 percent of synchronous at smallest roll diameter to 20 percent at greatest diameter.

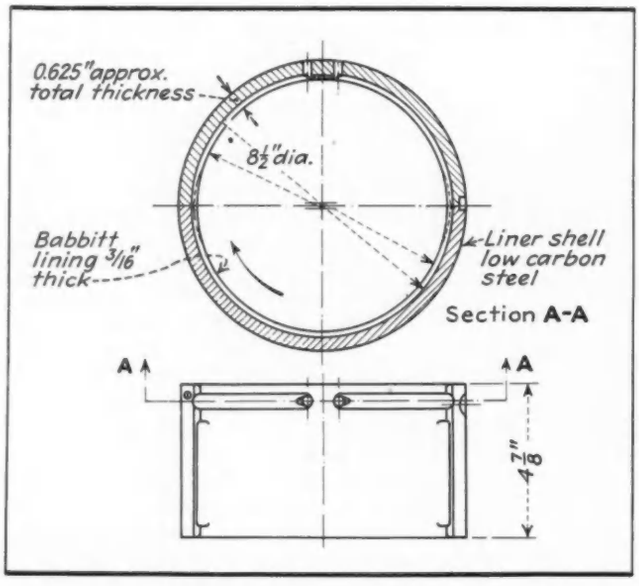
Special circuit connected through slip rings to rotor of winding motor combines resistance, inductance, and capacitance which determine rotor current according to voltage and slip frequency, in turn determined by rotor speed. These electrical components are balanced so as to affect speed-torque curve of the motor as shown in curve *A*. Unwinding motor is converted into magnetic brake by short-circuiting collector rings and feeding d.c. from copper-oxide rectifier into field windings. Horsepower characteristics are essentially constant, as shown in curve *B*.

Direction selector switch interchanges connections for winding and unwinding motors, thereby interchanging their functions and direction. The switch also reverses direction of nip roll drive motor. Time delay relays provide proper sequence of operations in starting and stopping the dyeing roll motors.

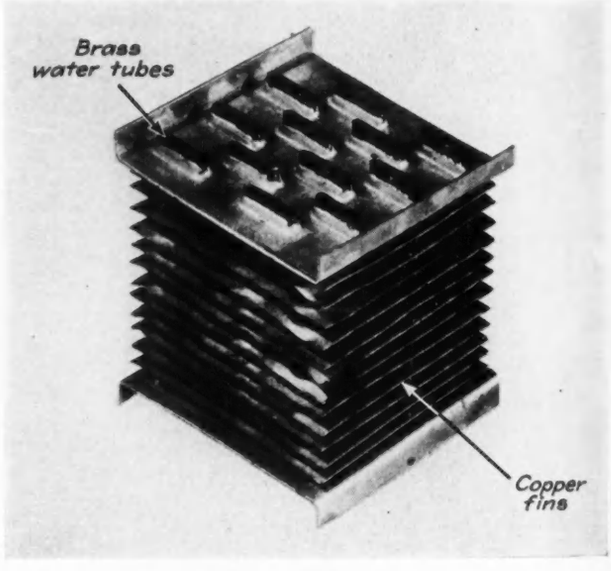
MODERN DESIGNS – Plastic Strip Held by Staples


Transparent vinyl acetate 0.020 in. thick
End rounded by cut-off shear
Extruded cellulose acetate
Flat wire staple
Lithographed 30-gage sheet steel, cemented in rabbet
Masonite 1/16 in. thick

Two problems had to be solved before extruded cellulose acetate end rails could successfully be used in place of extruded aluminum in this divider for VISIrecord equipment manufactured by Visible Index Corporation. Previously aluminum strips of same shape were crimped over edges of Masonite divider. First problem was to cut off plastics with neatly rounded ends from 8 ft. lengths, solved by special design of cut-off knives which simultaneously trim end of stock and end of piece being cut off. Similarly, development of special wire stapling head was necessary to obtain neat staples, which lie flush with the surface and penetrate halfway through the opposite leg of the extruded strip. Different widths of end rails, ranging from 0.14 to 0.28 in., are required to accommodate varying thicknesses of "build-up" of cards filed between dividers. Weight of end rails is reduced one-half. Unexpected advantage of plastics was reduced noise caused by dividers being slapped together by fast-working file clerks.



Large, thin-babbitt, steel-backed bearings are being designed by General Electric Company to replace thick-dovetailed cast babbitt bearings on large machinery. Typical is 8½ in. dia. elliptical bearing used on high speed turbines. Babbitt lining, centrifugally cast with strong bond to tinned surfaces of steel liner, is $\frac{3}{16}$ in. thick. This is more than is ordinarily required, but in this case thickness is determined as the minimum required to prevent scoring of the shaft should the bearing wipe out and let the shaft ride on the packing. If this should happen, the packing will call attention to the bearing failure before the blades are damaged or the shaft is scored. In smaller bearings General Electric is saving tin by switching from high-tin babbitts to alloys with tin content of around 1 percent. Typical bearing alloy is: antimony 14-15%, tin 1%, arsenic 1%, copper 0.25%, balance lead.



While structurally feasible, it is usually unsatisfactory to replace copper fins of radiators with brass in order to save some copper. Thermal conductivity of brass is so much lower than that of copper that increased thickness needed for the fins would result in use of more copper than pure copper fins would require. A radiator using steel fins and brass tubes would also require thicker steel fins and consequently a deeper radiator because the remaining air flow passages between the fins are decreased. The total increase in weight would be between 25 and 30 percent. An all-steel radiator would require kerosene as a coolant in order to prevent rusting. The lower specific heat of kerosene would make still more tubes necessary and increase the total weight by another 25 to 30 percent. Both types have been tried by the automotive industry and will work, but are regarded as last-ditch possibilities.

INDUCTION HEATING

For Hardening, Annealing, Brazing and Forging

H. B. OSBORN, JR.
Tocco Division, Ohio Crankshaft Company

WHEN A PIECE of metal is placed in the magnetic field produced by a high frequency electric current, hysteresis and the eddy currents generated in the metal produce heat. The rate of heat generation is roughly proportional to the frequency of the current and the wattage per unit of surface area of the power used as well as the proximity of the inductor to the surface. Frequencies from 2,000 to 10,000 cycles and upward of 100,000 cycles are being used extensively. An inherent characteristic of this phenomenon is that the currents induced in the metal are concentrated in a thin layer at the surface. The higher the frequency, other things being equal, the shallower will be the depth of heating.

The induction heater consists of some form of frequency changer, usually a motor generator set. For highest frequencies, such as of the order of 100,000 cycles, oscillating tubes are used. The equipment includes full automatic controls to regulate the heating cycle and all other operations of the machine.

In the equipment for surface hardening, the coil or inductor through which the current flows to set up the magnetic field may be one turn or several turns of copper tubing. The length and diameter of the inductor is made to suit the diameter and length of surface to be hardened, as shown in Fig. 1. When the surface to be hardened has reached the required temperature, quenching water is squirted through the inductor on the hot surface, thus producing the desired hardness. The quenching channel is an integral part of the inductor.

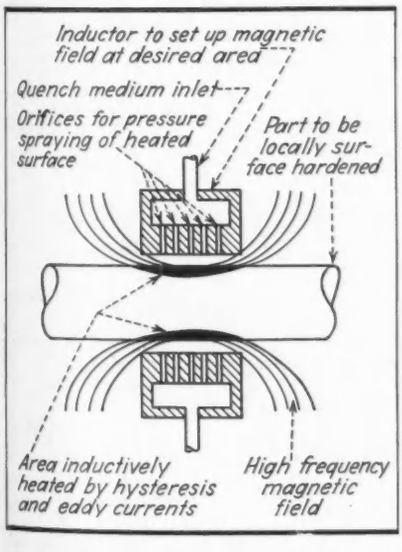

Inductor to set up magnetic field at desired area
Quench medium inlet
Orifices for pressure spraying of heated surface
Part to be locally surface hardened
Area inductively heated by hysteresis and eddy currents
High frequency magnetic field

Fig. 1—Diagram of an inductor block and magnetic field while heating a bar

TYPICAL APPLICATIONS OF INDUCTION HEATING

Bearing surfaces	Gear teeth	Valve stems	Surfaces that serve as
Worms for worm gears	Sprocket teeth	Tappets	roller or ball races,
Rocker arm shafts	Jaw clutches	Splines	as for needle bearings

OTHER APPLICATIONS

1. Brazing. Because of the rapidity of heating, oxidation is negligible and a controlled atmosphere is not required. An inherent characteristic of the induction heating process is that heat-treated parts can be brazed into assemblies without affecting physical properties obtained in prior heat-treatment.

2. Annealing and heat-treating of welds in assembled structures. Portable equipment is available, specially designed for this purpose.

3. Heating for forging operations. Absence of scale or heavy oxides is a distinct advantage because die life is increased.

DEPTH OF HARDENED CASE — Other things being equal, the higher the frequency the less will be the depth of hardness. Usually a depth of 3/32 in. is desirable for parts up to 2 in. or 3 in. dia., and up to ¼ in. case for larger diameters. The thinner the case the more the power required.

POWER REQUIRED—Other things being equal, the amperes per sq.in. of surface is a constant. The larger the diameter of the piece, the greater the voltage required.

SIZE LIMITATIONS—Today it is commercially practical by induction heating to harden bearings up to 12 in. in diameter with no limit on width since a progressive heating and quenching technique is employed. For larger diameters flame hardening may be more economical for low production items. However, there is no definite limit to the diameter of gear or chain wheel that is to have its teeth hardened. The teeth are heated and quenched one at a time and the gear is indexed automatically to the next tooth.

Frequencies of several hundred thousand cycles are available for surface hardening of small diameters. It would require a frequency of many hundred thousand cycles to confine the heat to the surface of a ⅛-in. dia. wire. For surface hardening ¼-in. dia. wire would require a frequency of several hundred thousand cycles; a ½-in. dia. bar would require approximately 10,000 cycles; and a bar of 1 in. dia. about 2,000 cycles minimum.

ADVANTAGES OF HARDENING BY INDUCTION HEATING—Because the heating cycle is extremely short, less than a second for some small pieces, there is practically no oxidizing of the surface. The area hardened can be localized as desired. The thickness of case is definitely controllable by automatic means. All the pieces hardened are uniform to an extremely high degree, unattainable by furnace heating. Skilled operators are not required.

DESIGN PRECAUTIONS—The usual precautions that apply in the design are the same as those which apply for any hardened part. An advantage of induction heating hardening is that because of the inherent characteristics of the process, corners such as at the roots of splines and at the base of collars or shoulders, do not get hot enough to harden thus assuring soft tough material where needed.

Because the hardening effects can be localized as desired, the designer can take liberties that are not permissible in furnace hardening.

WELDED STRAP ENDS

Simplified Design for Brake and Clutch Bands

GUSTAF JOHANSSON
Engineer, Link-Belt Speeder Corporation

COMPARED WITH the problems involved in the design of many large welded structures such as bases, frames, booms and dippers for electric shovels, the design of welded strap ends for brake and clutch bands entails little difficulty. Yet an analysis of the design of these simple parts presents a characteristic example of how welding can be utilized to advantage in the fabrication of many products.

Whether the welded structure be large or small, the design procedure is about the same. Each unit part is first studied to determine what deformation may take place under expected service loads, after which the welded joints are proportioned to obtain sufficient strength to carry the deforming forces. Economy in material, of course, is obtained by making every pound of material in the part carry a maximum safe load.

The particular example chosen for analysis in this article is a strap end for an 8 by $\frac{5}{16}$ in. brake band. In the original design the strap end was a steel casting, as shown in Fig. 1, which was fastened to the band by riveting. When judged on the basis of performance the part was entirely satisfactory, however, it was thought a welded design offered certain economies.

From a perfect riveted strap end which with counterbalancing members weighed 82.3 lb. that cost $11.35, a welded design weighing 14 lb., that cost $3.15 was finally arrived at.

This strap end, shown in Fig. 1, was designed to take the full load that could be transmitted by a 8x$\frac{5}{16}$-in. brake band, based on a tensile stress of 20,000 lb. per sq.in. and a shear stress of 12,000 lb. per sq.in. in band and strap, and 10,000 lb. per sq. in. shear stress for the rivets.

Net section of the band at section B-B is $(8 - 13/16) \times 5/16 = 7\,3/16 \times 5/16 = 2.25$ sq. in.

Total tensile strength of band is $20{,}000 \times 2.25 = 45{,}000$ lb.

Shear area of 8 − 3/4 in. rivets is $8 \times 0.5185 = 4.150$ sq. in.

Total strength of rivets in shear = 41,500 lb.

Section 3 1/4 in. back of center line of the pin hole is 2 1/2 in. wide × 4 1/8 in. deep. Area = 10.26 sq. in.

$$\text{Section modulus} = \frac{2.5 \times 4.125^2}{6} = 7.04$$

Center of gravity, the arm for a bending moment of this section is 2 7/32 in. outside of the center line of the band is $45{,}000 \times 2\,7/32 = 100{,}000$ in. lb.

The fibre stresses in this section are

$$F_1 \text{ (direct tension alone)} = \frac{45{,}000}{10.26} =$$

4,380 lb. per sq. in.

$$F_2 \text{ (for bending alone)} = \frac{45{,}000 \times 2\,7/32}{7.04} = 14{,}200 \text{ lb. per sq. in.}$$

Maximum combined tensile fibre stresses, $F = 18{,}580$ lb. per sq. in.

Because of stress concentration at *A*, the inner bend line, this tensile stress will be slightly increased.

Area of strap end at center of pin hole is $(2 \times 2\,7/8 - 3\,1/8)\,2.5 = 2\,5/8 \times 2.5 = 6.56$ sq. in.

With an allowable tensile stress of 18,000 lb. per sq. in., the minimum of area strap end at center of pin hole would be

$$\frac{45{,}000}{18{,}000} = 2.5 \text{ sq. in.}$$

Therefore, the design indicates a safe construction about the pin hole.

Bearing pressure on pin is

$$\frac{45{,}000}{3.125 \times 2.5} = 5{,}740 \text{ lb. per sq. in.}$$

which may be considered quite safe.

These calculations show a well balanced design with little waste of material and sufficiently strong through all sections.

The first design for a welded strap end is shown in Fig. 2. It looked rather good but failed at *A*, breaking at the end of the band, 4 inches back from center of pin hole. In this design it was assumed that a ¼-in. weld is good for 2,500 lb. per lineal inch. Total minimum length of weld required should be

$$\frac{45{,}000}{2{,}500} = 18 \text{ in.}$$

Actual length of weld is $2 \times (8 + 2.5) = 21$ in. which is safe.

Area of section of strap end where the break occurred $= 2\,1/2 \times 4 = 10$ sq. in.

$$\text{Section modulus} = \frac{2.5 \times 4^2}{6} = 6.66$$

Center of gravity of this section is 2 5/32 in. outside of the center line of the band causing a bending moment equal to $45{,}000 \times 2.156 = 97{,}200$ in. lb.

$$F_1 \text{ (direct tension alone} = \frac{45{,}000}{10} =$$

4,500 lb. per sq. in.

$$F_2 \text{ (bending alone)} = \frac{97{,}300}{6.66} =$$

14,650 lb. per sq. in.

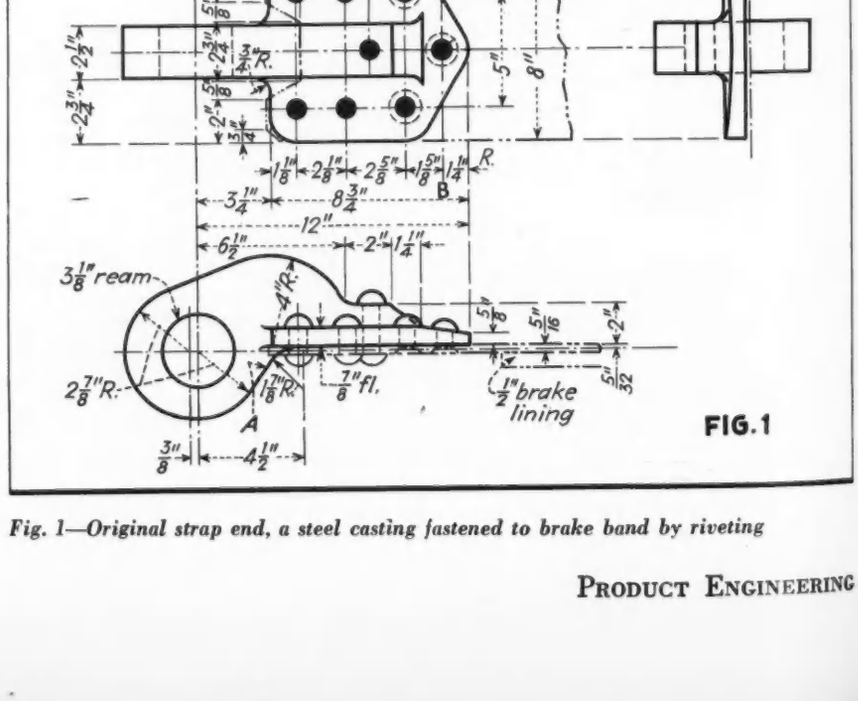


FIG. 1

Fig. 1—Original strap end, a steel casting fastened to brake band by riveting

Maximum combined tensile fibre stress, F = 19,150 lb. per sq. in.

This strap end broke because of stress concentration which was greater than anticipated.

The design shown in Fig. 3 is an improvement over that shown in Fig. 2, but it failed because of misplacement of two $\frac{9}{32}$-in. rivet holes which happened to be in line *A-A* as shown.

Assuming the strength of this band to be 45,000 pounds as before. Using ¼ in. weld, required length of weld is 18 in. at 2,500 lb. strength per lineal inch. The actual length of weld, however, was 24 in., which was well on the safe side.

Area of section of strap end 5 1/2 in. back of center of pin hole is 3 3/4 × 2 1/2 = 9.37 sq. in.

$$\text{Section modulus} = \frac{2.5 \times 3.75^2}{6} = 5.87$$

Center of gravity of this section is 7/8 in. outside of center line of band causing a bending moment of
45,000 × 7/8 = 39,300 in. lb.

F_1 (direct tension alone) = $\frac{45{,}000}{9.37}$ =
4,800 lb. per sq. in.

F_2 (bending alone) = $\frac{39{,}300}{5.87}$ =
6,700 lb. per sq. in.

Maximum combined tensile fibre stress F equals 11,500 lb. per sq. in., which is safe.

Strap end, shown in Fig. 3, was designed to take gradually the band load with a minimum of stress concentration. The band was coped back 4 in. and welded to strap end on both sides. The heel of the strap end at a distance of 2½ in. overlapped a full section of the brake band. The placement of two rivet holes with stress concentration in direct line with the cross weld caused the failure.

In Fig. 4 is shown the successful strap end. It is almost a duplication of the strap shown in Fig. 3, except for a more tapering tail end and that the rivet holes for the brake lining are in proper relation to the strap end, thereby preventing stress concentration at a point where stiffness is introduced by the added strap section.

The contemplated ideal design of a strap end, is shown in Fig. 5, in which all sections are balanced for strength with a concentric load condition. The views in Fig. 5 show in a simple way the principles of strain concentration.

Fig. 5(*A*) illustrates why protruding corners like those at *a*, *b* and *c* should be avoided. At first glance it may appear that these corners do no harm and that coping increases cost, however, on further thought it is obvious that such corners do harm the structure and that they are likely to cause excessive stress in the weld at *d* and start a crack. For just as a dent in a bar or shaft, or an inside sharp corner in a part, causes load concentration, so protruding corners cause bad concentrations of stress. When the band and strap end elongate under load, the triangular area *a-b-c* assumes a shape similar to *a-b-c*$_1$. To deform this corner, a certain amount of stress must be developed in this triangle with the result that unequal deforma-

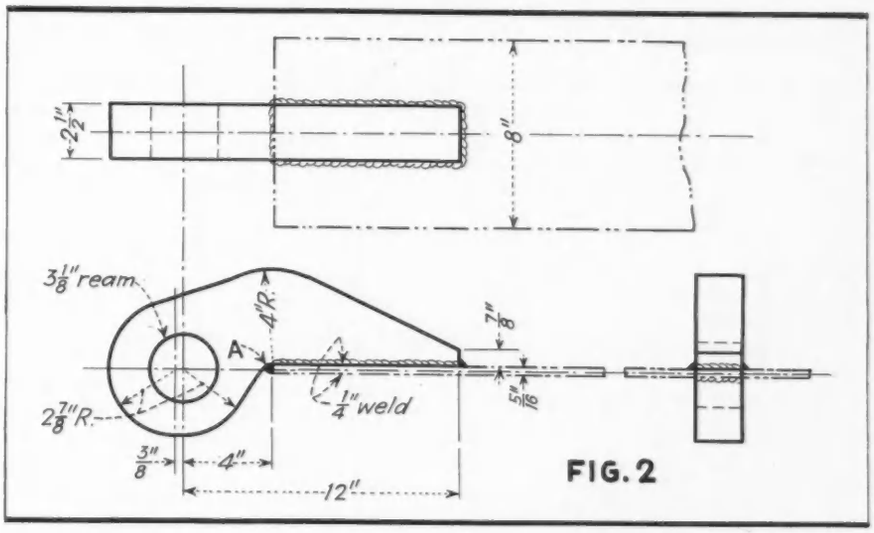


Fig. 2—First design of welded strap end failed at "A" at end of band

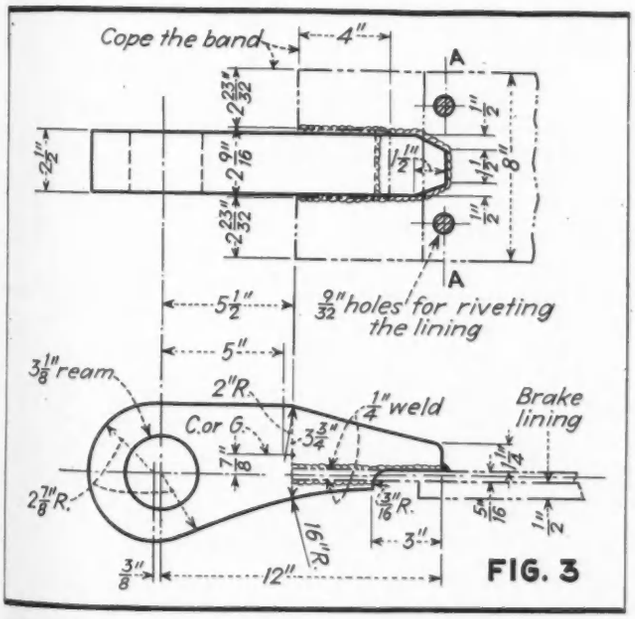


Fig. 3—Failure of this design was attributed to stress concentrations produced by rivet holes in line with weld

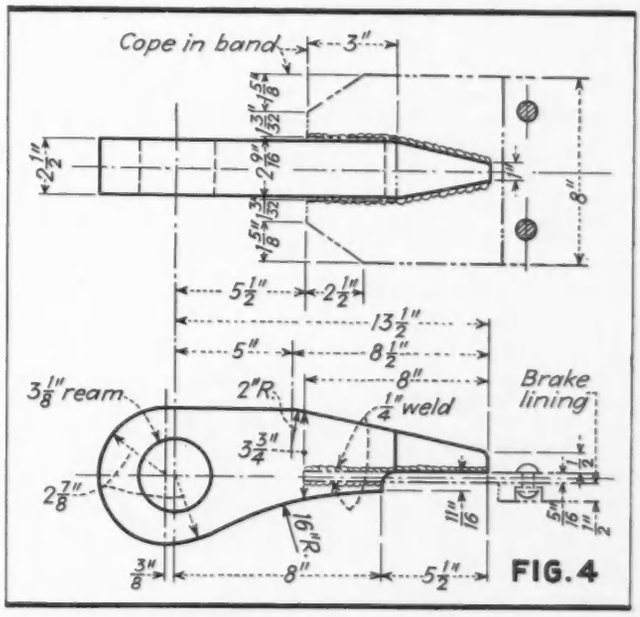


Fig. 4—A successful design which is almost a duplication of that shown in Fig. 3 except for position of rivet holes

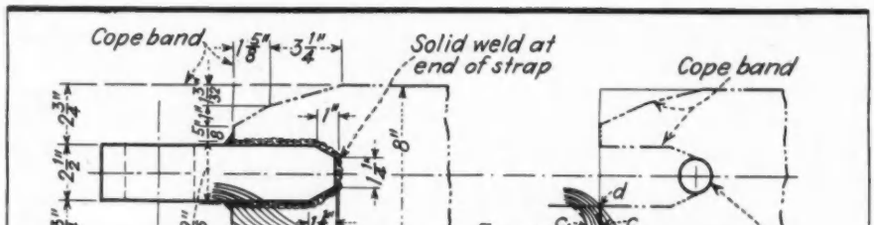


FIG.5

Fig. 5—Ideal design of a strap end in which all sections are balanced for strength

tion takes place at d where failure may occur. If the corner is removed, there is less strain at d and a more uniform flow of stress takes place as shown in the assembly view. In this design the material around the hole is also better proportioned.

The strap end shown in Fig. 5 is welded to an 8x$\frac{5}{16}$ in. band which has its section reduced by two $\frac{5}{32}$ in. rivet holes, so that the next cross section area is 2.33 square inches.

With allowable tensile force equal to 20,000 lb. per sq. in. the allowable load on this band is

$$20{,}000 \times 2.33 = 46{,}600 \text{ lb.}$$

Maximum shear between band and strap end equals 6,000 lb. per sq. in.

Minimum length of slot in band required is

$$\frac{46{,}600}{2 \times 6{,}000} = 3.88 \text{ in.}$$

Actual length equals 4 7/8 in.

Maximum allowable load per lineal in. of 1/4 in. weld equals 2,500 lb.

Required length of weld is

$$\frac{46{,}600}{2{,}500} = 18.6 \text{ in.}$$

Actual length of weld is

$4 \times 4\,7/8 + 2 \times 1\,1/4 = 19\,1/2 + 2\,1/2 = 22$ in.

Load per lineal inch is 46,600/22 or 2,120 lb.

To allow for stress concentration the section of strap end at end of band should be 60 percent larger than band proper, therefore, the required area is

$$2.33 \times 1.6 = 3.73 \text{ sq. in.}$$

Actual area $= 2 \times 2\,1/2 = 5$ sq. in., which is very safe.

Cross-section through the eye at center of hole, which should be 40 percent greater than band area, is $2.33 \times 1.4 = 3.26$ sq. in.

Required outside diameter is

$$\frac{3.26}{2.5} + 3.125 = 4.435 \text{ in.}$$

Actual diameter is 4.625 in.

Following good designing practice, the back section of the eye at Y has been increased $\frac{3}{8}$ in. in thickness above the cross section, from $\frac{3}{4}$ in. to $1\frac{1}{8}$ in. Thus this strap end has fully complied with good designing practice with no overload at any point and a minimum of concentrated secondary stresses.

Manufacturing costs for these five designs of strap ends are given in Table I. The welded strap ends may be made of cast steel or rolled steel, but because of the difference in cost rolled steel is generally used.

To increase further the strength of the strap end it should be cut and burned from 2½-in. thick bar, and the longitudinal line of the strap should be parallel with the rolled fiber in the bar; under no condition should it be cut crosswise. Because rolled steel is stronger along the grain than it is crosswise. This applies to rolled plates, of course, as well as all other rolled sections, therefore, the shop should be warned not to cut material crosswise if the part is to be subjected to large tensile forces. This is best done by making a cut diagram on the shop drawings.

Fillet welds along the sides of the strap longitudinal with the band are dependable even if all along this weld there should be an open space between strap and band at the center part of the band section, but under no condition should open spaces in the weld be allowed crossways the section. Such welds must be solid.

Designs shown in Figs. 6, 7 and 8 are three common, simple strap ends used on brakes and clutch bands. Strap end in Fig. 6 consists of a square or a round bar B welded to a hub C made either of a pipe spacer, or a round bar drilled for the pin, connected by welding to the band A. The proportions of the different parts should be in relation to the elastic limit of the material used. Should all the parts be of mild steel, which is well fitted for welding and of desired toughness, it is advisable to make the ratio of the different areas such that the areas of A, B and C are to each other as 1 to 1.20 to 1.40. For example, if t times w equals 1, area B should be 1.20 and D minus d times b should be 1.40 square inches.

Strap end shown in Fig. 7 is a flat bar burned out to shape, drilled for pin, then welded to band at one end. The net areas of the main sections should be

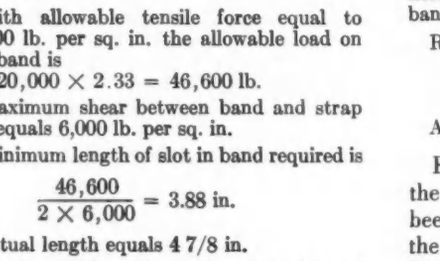


FIG. 6

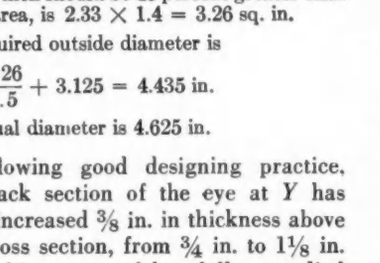


FIG.7

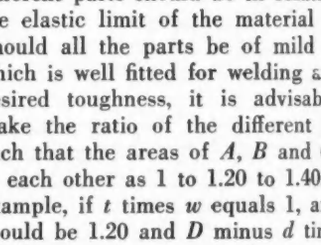


FIG. 8

Figs. 6, 7 and 8—Three common designs for strap ends used on brake and clutch bands

in same ratios as those for the preceding design.

Both the design shown in Fig. 6 and that shown in Fig. 7 have gradually tapering overlapping ends, thus avoiding too abrupt increases or decreases of sections. It also may be well to use light welding to prevent too much heat. In making strength calculations, of course, deduction should be made for holes in the band to receive the rivets which hold the lining.

The strap end shown in Fig. 8 is made by bending the band back upon itself and then welding. Notice how the welding is balanced to avoid eccentricity of stress distribution.

A design that uses two bars to form a strap end of forked construction is shown in Fig. 9. The two bars should be placed as indicated, in order to obtain a nearly uniform distribution of load across the band.

Design shown in Fig. 10 is for a wide band. Uneven contact of lining and drum surface is likely to occur when the parts are heated if the strap ends are not properly attached. For strength and stiffness, it is advisable to bend up the ends of the band as shown, thus forming a light cross beam which is able to take care of the pull in the outer edges of the band. With this construction the strain over the cross-section of the band is more uniform. Also, the strap ends pull straight and tangential to the circular friction surface.

Whether for brakes on automobiles, tractors, hoist machinery or what not, ordinarily the lining is riveted by brass or copper rivets to the steel band, and with each rivet a washer of similar material is used. Savings can be made in material, labor and life added to some types of lining which can be baked to the steel brake or clutch bands, thus omitting riveting. After cement is applied on the contacting steel and lining surfaces the parts are assembled with clamps and the cemented joint is baked at a temperature of approximately 380 deg. F. Electrically or gas heated ovens may be used for baking. To offset the expense of the equipment required, time and material savings will result such as:

A. A strip of steel in every brake and clutch band having a volume equal to the "thickness of the band times the reduction of rivet holes across the band times the full length of the band." This saving amounts to tons of steel now wasted, adding dead weight where not wanted;

B. Copper and other metals used in rivets and washers;

C. It is claimed that the baked lining can be worn almost down to the steel band, thus saving at least ⅓ of the lining;

D. All labor for marking up, drilling, setting up and riveting;

E. Less work on relining;

F. Also saving of scorched clutch and brake housings.

In concluding, the author feels that too much cannot be said on the subject of the elimination of areas of strain concentration and stress increases of high magnitude at points where a member with low fibre stress and no deformation connects to a highly stressed member with large deformation. Under shock loads, this condition may cause failure in a seemingly safe design.

Also, when weight is saved on one part it may also have effect on other parts by saving more weight, just as the light strap end reduces the needed weight for balance, thus causing less vibration and making other conditions still better throughout the machine.

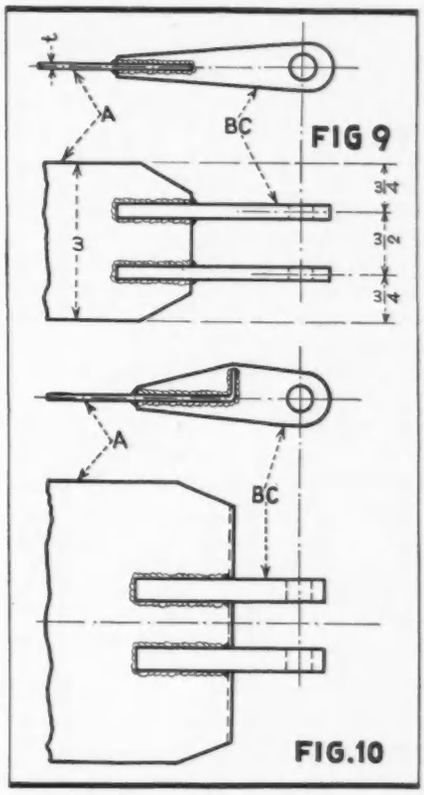
[EDITOR'S NOTE—Data and illustrations for this article were taken from a study submitted by the author to the James F. Lincoln Arc Welding Foundation.]

Table I—Comparative Data on Strap End Designs

	Riveted	Welded			
	Fig. 1	Fig. 2	Fig. 3	Fig. 4	Fig. 5
Pattern cost	Not included	None	None	None	None
Rough weight, lb	47	52½	51	58	27
Material and burning	Cast Steel	Rolled Steel	Rolled Steel	Rolled Steel	Rolled Steel
*Finished weight, lb	45	29	29	29¾	14
Grinding and dressing	.....	.....	.....	.....	.....
Layout work	.....	.....	.....	.....	.....
Drilling	.....	.....	.....	.....	.....
Rivets, in	7¾x2⅝ 1¾x3⅞	None	None	None	None
Riveting	.....	None	None	None	None
Coping of band	.....	None	.....	.....	.....
Drilling of band	.....	None	None	None	None
Welding rods	None	.....	.....	.....	.....
Welding	None	.....	.....	.....	.....
Reaming	.....	.....	.....	.....	.....
Oxygen	None	.....	.....	.....	.....
Acetylene	None	.....	.....	.....	.....
*Additional counterweight to balance strap end, lb	37.3	24	24	24¼	0
Total finished weight, lb	82.3	53.0	53	54	14
Credit for scrap iron, lb	2	23½	22	28½	13
Total cost incl. material, labor, current, overhead, etc., but no pattern cost	$11.35	.....	.....	$5.74	$3.15
Weight Ratio, percent	587.0	379.0	379.0	385.0	100
Cost Ratio, percent	361.0	.....	.....	182.5	100

*Weights indicated here are added to obtain the total finished weight required to make up a perfect working strap end and a well balanced clutch.

Note: Cost figures vary for every shop, and for this reason no detail cost is given, only the final, actual cost which should be relatively right for all shops.



Figs. 9 and 10—Welded strap ends of the fork or clevis type

SELENIUM RECTIFIERS – II

Characteristics, Applications and Design Factors

CAROLE A. CLARKE
International Telephone & Radio Manufacturing Corporation

SELENIUM rectifier plates inherently do not require cooling fins because of adequate cooling surface. However, such fins are often used to extend the current capacity of the largest size of plate as well as to obtain current values intermediate between the normal ratings of smaller plates.

Basic types of selenium rectifiers and their ratings are listed in Table I. The normal full load output current in amperes is given for each type on the basis of one plate (or several in series) in each branch of the listed rectifier circuits. Output currents can be increased by the connection of plates in parallel. Plate reverse voltages, outside diameters and d.c. valve ratings are included in the table. The forward currents permissible for d.c. valve use are higher than the equivalent half wave rating for a.c. operation because there is no simultaneous heating due to reverse current and the form factor is unity. The maximum permissible d.c. reverse potential is less than the r.m.s. a.c. values.

With fan cooling at an air velocity of 55 ft. per min. Type 7, for example, gives 150 percent of the convection cooled current ratings listed in Table I. If the air velocity is increased to 90 ft. per min. the ratings are twice the values in the table and at 120 ft. per min. 2½ times the listed values.

In oil at 35 deg. C., the Type 4 selenium rectifier gives 167 percent of the currents indicated in the table.

Along with unusual stability of operating characteristics, the good voltage regulation of the selenium rectifier results from its low forward resistance. Regulation from no load to full load for a single-phase bridge rectifier with a resistive load is of the order of 15 percent. The addition of fins, or wide spacing, for obtaining increased current density may increase this value by about 5 percent; with highly intermittent use a further increase may occur if too small a rectifier is selected. The largest portion of the voltage change occurs from zero output current to some small value of current of the order of 20 percent of full load. From there on the voltage change is almost linear and the regulation from, say, half load to full load, is only about 6 percent. The use of a small additional load or bleeder resistance improves the regulation by preventing the output current from decreasing to zero.

Where close regulation is essential various regulating devices may be em-

Table I—Sizes and Current Capacity of Selenium Rectifiers

Type No.	Diameter In.	Max. R. M. S. A.C. Volts Reverse Per Plate	*Max. Continuous D.C. Amperes for One Plate Per Arm at 35 Deg. C. Ambient, Inductive or Resistive Load						Max. Continuous D.C. Rating as Valve at 35 Deg. C. Ambient		Max. No. of Plates Per Stack	Using Plate No.
			Single-Phase			Three-Phase			Amperes	Blocking Voltage		
			Half Wave	Bridge	Center Tap	Half Wave	Bridge	Center Tap				
1	¾	18	0.040	0.075	0.075	0.100	0.110	0.130	0.06	15	36	1
2	1	18	0.075	0.150	0.150	0.200	0.225	0.270	0.12	15	36	2
20	1	18	0.11	0.22	0.22	0.29	0.33	0.4	0.17	15	28	2
3	1⅜	18	0.15	0.30	0.30	0.40	0.45	0.55	0.23	15	36	3
21	1⅜	18	0.23	0.45	0.45	0.6	0.67	0.82	0.34	15	28	3
4	1¾	18	0.30	0.60	0.60	0.80	0.90	1.1	0.45	15	40	4
10	1¾	18	0.39	0.78	0.78	1.0	1.1	1.4	0.58	15	28	4
9	2⅝	18	0.58	1.1	1.1	1.5	1.7	2.1	0.87	15	28	4
5	2⅝	18	0.60	1.2	1.2	1.6	1.8	2.2	0.90	15	40	5
11	2⅝	18	0.78	1.6	1.6	2.1	2.3	2.8	1.2	15	28	5
12	3⅜	18	0.90	1.8	1.8	2.4	2.7	3.3	1.4	15	28	5
13	4⅜	18	1.1	2.2	2.2	2.9	3.3	4.0	1.7	15	28	5
6	3⅜	16	1.2	2.4	2.4	3.2	3.6	4.5	1.8	12	40	6
14	3⅜	16	1.5	3.1	3.1	4.1	4.6	5.8	2.4	12	28	6
15	4⅜	16	1.8	3.5	3.5	4.6	5.2	6.5	2.7	12	28	6
16	4⅜	16	1.9	3.8	3.8	5.0	5.6	7.0	2.9	12	24	6
17	6″ sq.	16	2.7	5.4	5.4	7.2	8.1	10.0	4.1	12	28	6
7	4⅜	14	2.0	4.0	4.0	5.3	6.0	7.5	3.1	12	40	7
18	4⅜	14	2.6	5.2	5.2	6.9	7.8	9.7	4.0	12	28	7
19	6″ sq.	14	3.7	7.4	7.4	9.8	11.1	13.3	5.7	12	28	7
8	8″ sq.	14	5.0	10.0	10.0	13.0	15.0	18.0	7.5	12	28	7

* These ratings do not apply to battery charging or condenser loads.

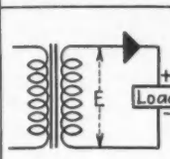
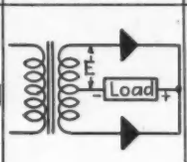
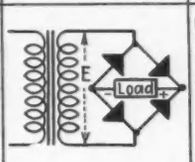
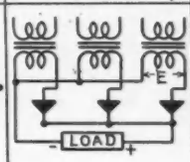
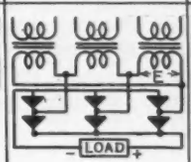
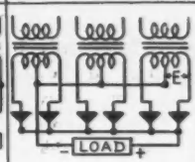
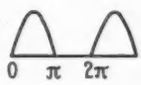
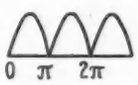
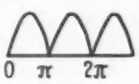
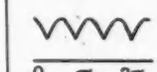
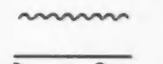
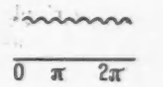
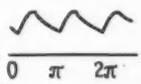
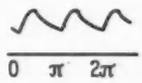
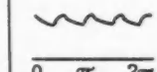
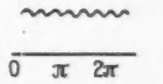
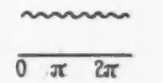
CIRCUIT	SINGLE-PHASE HALF WAVE	SINGLE-PHASE CENTER TAP	SINGLE-PHASE BRIDGE	3-PHASE HALF WAVE	3-PHASE BRIDGE	3-PHASE CENTER TAP
RECTIFIER CONNECTION						
RESISTIVE LOAD Approximate output voltage wave shape						
CAPACITIVE LOAD Approximate output voltage wave shape						
NUMBER OF PLATES IN SERIES E=AC RMS volts V_p=Rated volts per plate	RESISTIVE LOAD $\frac{E}{V_p}$	RESISTIVE LOAD $\frac{2E}{V_p}$	RESISTIVE LOAD $\frac{E}{V_p}$	RESISTIVE LOAD $\frac{\sqrt{3}\,E}{V_p}$	RESISTIVE LOAD $\frac{E}{V_p}$	RESISTIVE LOAD $\frac{2E}{V_p}$
	CAPACITIVE LOAD $\frac{(1+\sqrt{2})\,E}{V_p}$	CAPACITIVE LOAD $\frac{2\sqrt{2}\,E}{V_p}$	CAPACITIVE LOAD $\frac{\sqrt{2}\,E}{V_p}$	CAPACITIVE LOAD $\frac{(1+\sqrt{2})\,E}{V_p}$	CAPACITIVE LOAD $\frac{\sqrt{2}\,E}{V_p}$	CAPACITIVE LOAD $\frac{2\sqrt{2}\,E}{V_p}$
THEORETICAL RIPPLE	121 percent	48.3 percent	48.2 percent	18.3 percent	4.2 percent	4.2 percent
MAXIMUM THEORETICAL OUTPUT EFFICIENCY $\frac{E_0\,DC\ \ I_0\,DC}{E_0\,AC\ \ I_0\,AC}$	40.5 percent	81.1 percent	81.1 percent	96.8 percent	99.8 percent	99.8 percent

Fig. 10—Basic characteristics of rectifier circuits

ployed, such as the saturable core reactor type of regulating rectifier which is widely used for telephone central office power supply. A tapped or continuously variable transformer may be used when compensation for line voltage fluctuations is desired or variation required in output voltage.

In order to take into account the various form factors which differ considerably with various types of load and rectifier arrangements, the efficiency of the selenium rectifier is considered to be its volt-ampere efficiency, obtained by a wattmeter measurement of input and a moving-coil voltmeter and ammeter measurement of output. Such measurements show the rectifier efficiency between 65 and 85 percent, depending on the nature of the circuit and the load. These efficiencies apply to convection cooled rectifiers without cooling fins and operation at maximum voltage. The high efficiency is maintained with remarkable uniformity from about 20 to 150 percent of full load. The selenium rectifier differs in this respect from most electrical equipment which normally reaches maximum efficiency only in the neighborhood of full-load current. Efficiency of the rectifier, however, is dependent on the operating voltage per plate. The aim of the designer is, therefore, to provide an adequate but not excessive number of plates for each specific application.

In comparison with tube rectifying equipment, the selenium rectifier is usually found to have a higher efficiency for outputs of any current at voltages up to about 80 to 120 volts. Tube equipment ordinarily has a constant internal drop of about 15 volts, regardless of output voltage, whereas the total internal drop of a selenium rectifier must rise as its voltage and hence the required number of plates is increased. Furthermore, tube efficiency is lowered by the filament or heater losses. The efficiency of the selenium rectifier, therefore, is higher in the range within which the loss due to its internal drop is less than the total losses of the tube rectifier.

Rotating machinery, such as the motor generator, compared with the selenium rectifier, is about equally efficient at full load but its efficiency drops off rapidly as the load deviates from full load.

Choice of circuit schemes ordinarily may be left to the rectifier designer. Under certain conditions advantage should be taken of the fact that three-phase circuits have a smaller ripple then single-phase rectifier circuits and their higher frequency is easier to filter. Inasmuch as the utility factors for three-phase as compared with single-phase circuits are more favorable, the resultant three-phase rectifiers are smaller and less expensive.

In Fig. 10 are listed the basic characteristics of the six most common rectifier circuits. The single-phase half wave circuit is ordinarily not economical except for low voltage and current output since only half of the cycle is utilized. The use of a condenser across the load maintains some current flow during the half-cycle which is suppressed by the rectifier. Rectifier reverse current flows through the load.

The single-phase full wave bridge arrangement is perhaps most widely used. The ripple voltage has twice the frequency of the a.c. supply and the rectifier back current does not flow through the load.

The single-phase full wave center tap arrangement provides the same performance as the bridge circuit but is only half as large. It is applicable to output voltages of the order of 6 volts or less and requires a center tapped transformer. For higher output voltages the number of plates required will be the same as for the bridge connection and it is then more economical to use

the latter circuit since it does not require a center tapped transformer.

The three-phase half wave circuit differs from the corresponding single-phase arrangement in that current flows through the load throughout the entire cycle due to overlapping of the output from the three phases.

The three-phase bridge circuit is one of the most useful for large power outputs; it has the highest efficiency and the lowest ripple voltage.

The three-phase center tap circuit is similar in connection to the single-phase equivalent. It offers the same performance as the three-phase bridge connection but has a lower voltage and higher current output per plate.

In Fig. 11 are given the approximate current factors in these rectifier circuits, especially useful in the determination of wire and bus bar sizes.

In order to design a rectifier for a specific application, an outline of the limiting conditions is required. A circuit schematic including constituent electrical constants conveys the clearest definition of the operating requirements and frequently enables the designer to suggest alternate arrangements warranting consideration. Actual performance requirements and details of service conditions are preferable to the inclusion of arbitrary safety factors.

Obvious requirements include d.c. output voltage and current. Stated tolerances for each aid the designer, whose choice is limited to fixed current and voltage steps. First cost usually is a function of rectifier output and varies with the number of plates required, making it desirable to hold size to a minimum. The number of phases of the power supply is important but the frequency is relatively unimportant except for capacitive loads or smoothing filter circuits inasmuch as the rectifier ratings hold for frequencies up to 1,000 cycles per second. Voltage variation of the a.c. supply line should be stated.

Upper and lower ambient temperature limits must be known as well as the duty cycle, that is, whether continuous or intermittent. If the latter, details of current changes are required together with the length of time each value is to be maintained, as well as the length of each "off" period. Necessary also are voltage regulation limits between stated loads and details of switching, that is, whether on the line or load side of the rectifier.

The nature of the load is an important design factor. Inductive and resistive loads are usually treated similarly but capacitive loads and battery charging service require different designs. For capacitive loads, the value of the capacity and the supply line frequency must be known in order to determine the a.c. input to the rectifier. If a filter is used, it is considered as an inductive or capacitive load according to whether its input section comprises a choke coil or a condenser.

The a.c. voltage required for a given d.c. output voltage depends on the circuit and load conditions. For a resistive or inductive load, the r.m.s. input voltage of a single-phase bridge rectifier is of the order of 30 percent above the mean output voltage; for a three-phase bridge rectifier, the input voltage is about 12 percent below the mean output voltage. Similarly, for a single-phase, half wave rectifier the input voltage is about 180 percent above the output; and, for a three-phase half wave rectifier it is about 8 percent above the output.

In addition to stated line voltage, ambient temperature and other known variables, the rectifier designer must consider full compensation for aging as well as allowances for variation in ventilating conditions.

The housing for the complete equipment should preferably be small in horizontal cross-sectional area and as high as practicable in order to enhance "chimney" action for convection cooling.

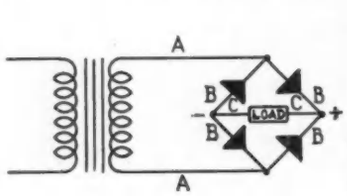
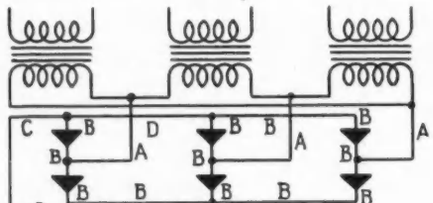
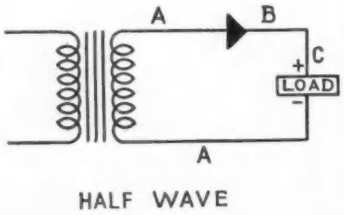
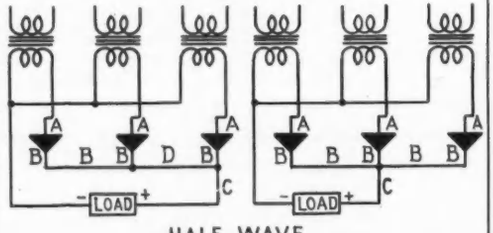
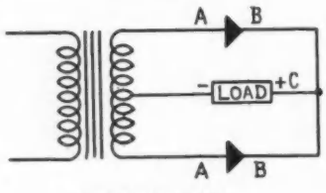
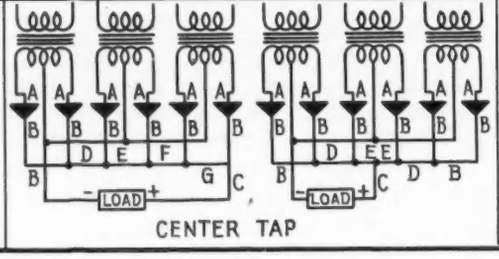
SINGLE-PHASE				3-PHASE			
	SECTION	RESISTANCE LOAD FACTOR	BATTERY LOAD FACTOR		SECTION	RESISTANCE LOAD FACTOR	BATTERY LOAD FACTOR
 BRIDGE	A	1.15	1.7	 BRIDGE	A	0.85	0.86
	B	0.80	1.2		B	0.60	0.61
	C	1.15	1.6		C	1.0	1.02
					D	0.85	0.86
 HALF WAVE	A	1.80	2.5	 HALF WAVE	A	0.65	0.70
	B	1.80	2.5		B	0.65	0.70
	C	1.80	2.5		C	1.06	1.08
					D	0.86	0.90
 CENTER TAP	A	0.80	1.2	 CENTER TAP	A	0.46	0.48
	B	0.80	1.2		B	0.46	0.48
	C	1.15	1.6		C	1.00	1.02
					D	0.62	0.64
					E	0.75	0.77
					F	0.85	0.87
					G	0.95	0.97

Fig. 11—Approximate current factors in rectifier circuits. To obtain effective current (a.c.) in any section, multiply the d.c. or average value of the rectified output current by the appropriate factor

INDUSTRIAL PLASTIC PARTS

General Design Considerations

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THE MORE the product engineer learns about design of industrial plastic parts, the less time and trouble it will take to go into production with a new design, and the more efficient will be his use of plastic materials. Consider the creation of a molded part. Conception takes place in the mind of a product engineer, the design is roughed out, and sketches are relayed via a molder's salesman to the plastics engineer. A price quotation is prepared through joint collaboration of engineering, tool and manufacturing representatives, each offering constructive criticism which gradually perfects the design.

Should the quotation be favorably received, the customer places an order. Final details can best be settled with a model. The mold is then designed, built, sampled and checked for accuracy —a process taking several weeks and often running into months for complicated jobs. Once the customer approves samples, the problem is reduced to that of molding enough parts.

Selecting the Material

The first serious consideration of a new plastic part after a rough plan has been made must settle the question of compound. Industrial applications of plastics make demands covering all imaginable characteristics of materials. For example, electrical insulation must satisfy requirements for insulation, dielectric strength, power factor, dielectric constant, arc resistance, moisture absorption, heat resistance and mechanical strength. Mechanical strength divides into impact, flexural, tensile and compressive strengths which in turn are limited by their respective fatigue values. Other molded parts must excel in qualities like weathering resistance, machinability, light weight, abrasion resistance, low coefficient of friction, and light transmission.

Selection of the material then involves a careful study of all plastic materials available, followed by a matching of their properties with the requirements of the job—a subject outside the scope of this article but amply covered in other articles which have been published in PRODUCT ENGINEERING. Occasionally no amount of part redesigning will overcome some deficiency in materials where a particular combination of qualities is required. Should the job warrant, laboratory facilities are available to develop new compounds combining necessary features. Failing this the customer is told frankly that he must look elsewhere.

Designing the Part

Having chosen a compound, engineers submit the roughly designed part to a scrutiny for changes necessary to good molding. "Can we extract the part in one piece?", is sometimes awkward to answer. Loose wedges which come out of the cavity with each molded part make possible many types of undercuts. Two-ram angle presses give freedom for undercuts in directions at right angles to each other. Some parts will be cheaper to mold solid and machine afterward; some parts should be made in two or more pieces and assembled afterward.

Once the problem of extraction is settled, dimensions take the center of attention. Items of wall thickness, draft, location and size of holes, tolerances, volume, and cross-section area perpendicular to molding direction are considered from the viewpoints of mold construction, ease of molding, probable sources of scrap and ultimate satisfaction to the customer.

Wall thickness must in all cases give sufficient mechanical strength or the part is useless. Both the customer and plastics molder should satisfy themselves beforehand by suitable stress calculations. From the molder's point of view a uniformly thin wall section is desirable for both thermoplastic and thermosetting compounds. Thermoplastic materials will show less tendency toward shrink marks as well as requiring smaller volumes of relatively expensive material. Thermosetting materials will cure evenly and with fewer shrinkage strains caused by large variations in cross-section. Rate of production for both materials is determined to a large degree by the thickness of the largest section. Generally accepted minimum

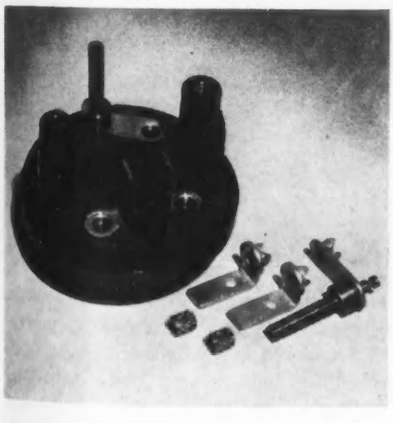


Fig. 1—Inserts of many sizes and shapes can be molded into plastic parts, as this switch-gear part illustrates

Table I—Location and Size of Molded Holes

Minimum Hole Diameter In.	Maximum Hole Length, In.		Minimum Side Wall, In.	
	Blind Hole	Straight through	Hot molded	Cold molded
1/16	1/16	1/8	1/16	3/32
5/64	3/32	3/16	1/16	3/32
3/32	1/8	1/4	1/16	3/32
7/64	5/32	5/16	3/32	3/32
1/8	3/16	3/8	3/32	1/8
5/32	1/4	1/2	3/32	1/8
3/16	5/16	5/8	1/8	1/8
7/32	3/8	3/4	1/8	3/16
1/4	7/16	7/8	1/8	3/16
5/16	9/16	1 1/8	5/32	1/4
3/8	11/16	1 3/8	5/32	1/4
7/16	13/16	1 5/8	3/16	5/16
1/2	15/16	1 7/8	3/16	3/8

wall thickness is $\frac{3}{16}$ in. for the small general purpose part. For larger pieces (radio cabinets, housings) wall thickness should be $\frac{1}{4}$ to $\frac{3}{8}$ in. A domed top for molded parts increases strength and facilitates molding.

As in molding other materials, it is customary to allow taper on the cavity walls in order to ease extraction of the molded part. A quarter degree per side is usually sufficient for hot molded compounds while the more abrasive cold molded compounds will require twice as much. Occasionally straight sides are required on certain faces of a molded part. Flexibility of the newly molded part usually permits extraction providing all other faces have adequate taper. High bosses or deep recesses should have 5 deg. taper with generous radii at top and bottom.

A guide to the location and size of molded holes is given in Table I. Straight-through molded holes may be longer than blind holes because the molding pins are supported on both ends. Many designs calling for long through-holes can be changed so that the small diameter is molded a short distance and the rest of the hole is a larger diameter, thus obtaining necessary mold pin strength. While Table I is to be used as a guide, favorable molding conditions such as occur in injection and transfer molding allow many exceptions.

Drilled holes offer a good alternative for difficult molded holes. Punched holes are usually limited to relatively thin laminated plastics and allow an innumerable variety of shapes in addition to round holes. Maximum thickness of laminated plastics that can be punched are:

Type	Hot	Cold
Paper	$\frac{3}{16}$ in.	$\frac{1}{8}$ in.
Fabric	$\frac{1}{4}$ in.	$\frac{3}{16}$ in.

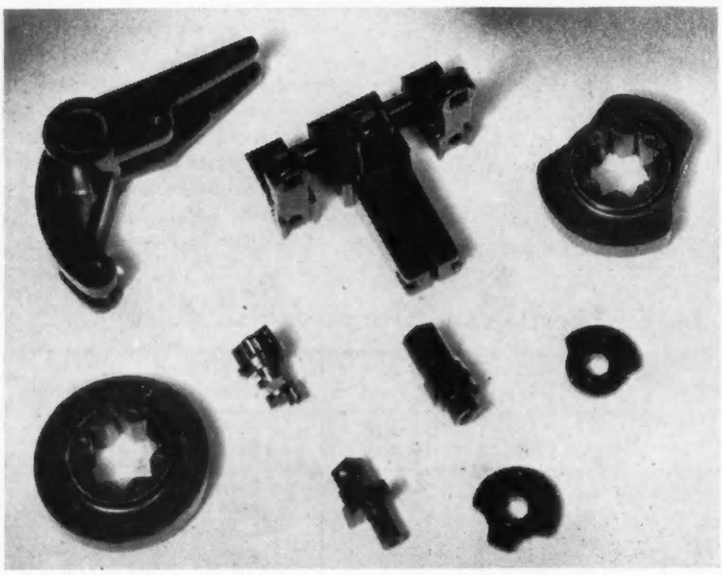
Unnecessarily close tolerances spoil many otherwise good molding jobs. Table II lists tolerances which can be expected of the average hot or cold molded part in a direction perpendicular to applied pressure. Tolerances in the direction of molding vary with the type of mold. For hot compression molding 0.005 in. to 0.020 in. build-up can be expected on the parting line. Small molds, simple parts, and free flowing compounds are conducive to thin cutoff. The reverse is true for complex parts and low flow materials. Transfer and injection molds which are closed when the compound enters can usually hold less than 0.005 in. build-up thus approximating tolerances of Table II for all directions. Even these tolerances may not be close enough and parts may require a final machining to size.

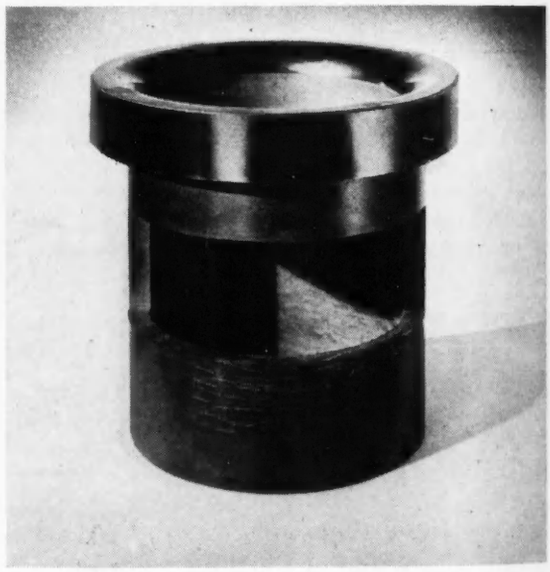
Table II—Tolerances of Molded Plastics

Nominal Dimensions In.	Tolerances Plus or Minus, In.	
	Hot Molded	Cold Molded
0.5 or less....	0.002	0.010
0.500 to 2.000.	0.005	0.013
2.000 or over..	0.0025 per in.	0.10 and 0.003 per in.

Many factors are responsible for the relatively large tolerances used in the plastics industry. Molds must be made somewhat oversize to allow for shrinkage, both chemical and thermal. Different batches of the same compound may have slightly different shrinkages. Variations in the length of cure with the same batch will change shrinkage. Uneven rates of cooling can permanently warp otherwise perfect parts. The daily changes in relative humidity and temperature will affect size. All these items can only be imperfectly controlled, hence extra close tolerances should not be specified if avoidable.

Total volume is always calculated and converted to weight of compound. Fair accuracy is necessary since compound prices per pound range from 15 to 85 cents, and a few hundred thousand parts are often at stake. Volume is particularly important for injection molded parts, for not only is the compound expensive, but the pressing machines must have the capacity to plasticize and inject the required compound in a rea-





Mechanical properties of high strength, light weight, resilience, shock resistance, toughness and easy machinability of laminated, cotton-flock molded and other industrial plastic compounds leads to their selection for many parts such as silent gears and cams, rocker arms, rollers, levers, pump impellers, machine housings, load carrying panels. Typical examples are small, high-impact, light-weight molded machine parts at left. Water-lubricated roll neck bearing (right) for seamless steel tube mill is fabric-filled phenolic, molded to 60-deg. shape—an outstanding new development. Such bearings outlive metal bearings by several times, withstand mechanical shock and high compressive loads, and absorb less power than metal bearings. Starting resistance may be greater, and continuous lubrication is essential to dissipate heat. Compound can be impregnated with graphite to reduce friction and starting torque.

sonably short cycle of the machine.

Cross-sectional area of the molded part perpendicular to the direction of molding determines the size of molding press. Clamping pressure per square inch of molding area runs from 3,000 to 8,000 lb. per sq. in. for compression molding and 15,000 to 30,000 lb. per sq. in. for injection molding.

Intimately associated with dimensions is the use of radii on corners. Such "streamlining" of the molded part assists flow of material in the cavity and produces a strong part. Especially is this true for high impact materials using long fibre fillers. The use of fillets reduces stress concentrations in both the molded part and the mold. Mold construction is simplified by eliminating square corners which have to be dug out by hand and there are no sharp corners on plungers to wear round.

An exception to general fillets is a square edge located on the flash line of the molded part. Mold construction is simplified and cleaning of the molded part is cheaper and neater.

Strengthening bosses and ribs are used to assist flow of compound and render parts more rigid to warping. Frequently thick sections in a proposed part are redesigned to thin sections plus ribs, thus saving compound and shortening the molding cycle.

Relatively high coefficient of thermal expansion of plastics is also an important design consideration. Designs where plastics are used in conjunction with other materials, such as in aircraft windows, should allow for unequal expansion at the joint.

Inserts and Threads

A large number of molded parts make use of inserts. Many materials have been tried, but metals, glass, and other molded plastic parts are most common. Reasons for inserts are excellent. Simplified assembly problems reduce cost, reinforcement lends strength, tolerances are closer, arc-resistant inserts overcome compound deficiencies, corrosion-resistant plastics protect metal inserts, current carrying inserts can be well insulated and supported, and inexpensive roughly finished inserts can be made attractive.

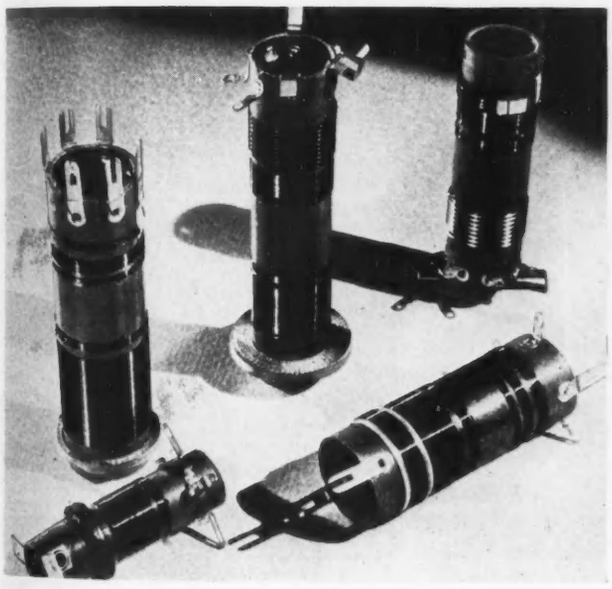
Successfully designed inserts have adequate anchorage within the part. In the case of round metal inserts, knurling may be sufficient. Grooves cut around hexagonal inserts provide anchorage against pulling out while the angular perimeter prevents turning. Many inserts have such odd shapes that they require no extra anchorage.

In all cases the compound covering the insert should be sufficiently thick. A good general rule for thickness of compound around inserts is one-half the diameter of the insert. Compound usually must support any mechanical load on the insert and such stresses should have a high safety factor. Also, the difference in coefficient of expansion between the compound and insert causes the compound to shrink tightly around most metal inserts. For thermosetting materials extra large inserts may set up in the plastic breaking strains which an annealing oil bath (in neighborhood of 115 deg. C) can effectively relieve. Thermoplastic materials have been used in thicknesses of $\frac{1}{8}$ to $\frac{1}{8}$ in. on metal panels of considerable size. For such parts cold flow takes care of the difference in coefficient of expansion.

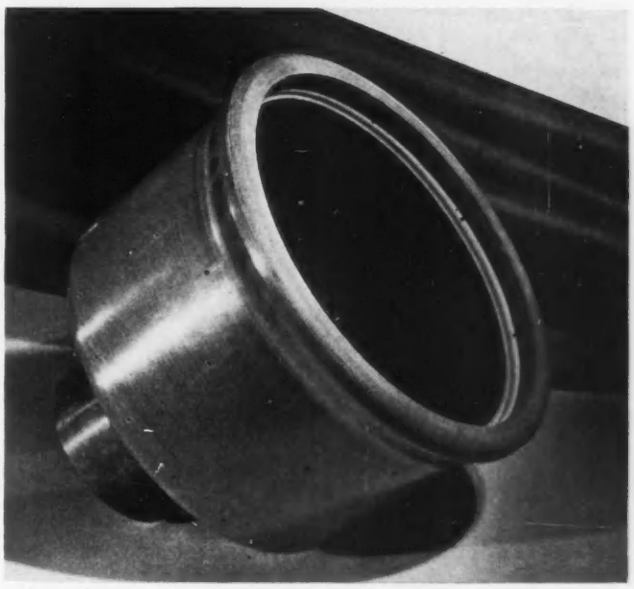
Another essential of inserts is to retain shape during molding. The type of mold and the method of insert support will govern strength requirements. Injection and transfer molds, in which compound pressure builds up more or less evenly in all directions, can use quite delicate inserts. Flat glass plates held in place by vacuum have been molded into radio receiver escutcheons by injection methods.

When inserts cannot conveniently be located in the cavity during molding, they sometimes can be placed in molded holes in the hot and still plastic parts immediately after parts are taken from the molds. Contraction of the plastic upon cooling holds the inserts firmly.

Few relatively complex plastic parts are designed without threaded sections. The simplest designs require tapped holes which have either molded threads, machined threads or tapped metal in-



Electrical applications are typified by laminated radio coil forms where desired properties are dielectric strength, low loss factor and low water absorption. Molded phenolics and ureas have been used for years in countless small electrical parts. Laminated fabric and paper have typical uses in instrument panels, insulation strips. Heat and arc-resistant cold molded plastics have uses in field rheostats for motors and generators. Polystyrene is gaining in electrical uses.



Chemical resistance of plastics is a deciding factor in many industrial applications, typical of which is this molded-laminated rayon spinning bucket. Plastics are available which resist action of various acids, salts, alkalies, oils, alcohols and solvents, but no single plastic will resist all. Materials with low water absorption maintain good electrical properties and weather well. Typical chemical applications include tank linings, filtering equipment, food machinery.

serts. Molded threads require unscrewing of the part from the pin or vice versa which is apt to be a slower process than either machining or the use of inserts. However, power driven unscrewing devices have been developed which, though expensive, speed production. Machined threads for tapped holes are readily made using either drilled or molded tapping holes and a reversible chuck. This type of service is particularly hard on taps. Tapped metal inserts made on automatic screw machines give threads which usually require no more operations or at the most a simple chasing to remove excess molding compound.

The most satisfactory type of molded external thread is made in a ring which must be unscrewed. Split wedges are also used, but these leave a parting line across the threads. As the mold wears the flash at the parting line becomes thicker and the thread diameter goes out of round. Very shallow molded threads can sometimes be stripped from the cavity without unscrewing.

Analyzing Mold Costs

Molds are broadly classified as compression, transfer, injection and extrusion. Thermosetting compounds are commonly used with compression and transfer molds, while thermoplastic materials find greatest application in injection and extrusion equipment. Thus, choice of compound is usually the most important item in determining mold type.

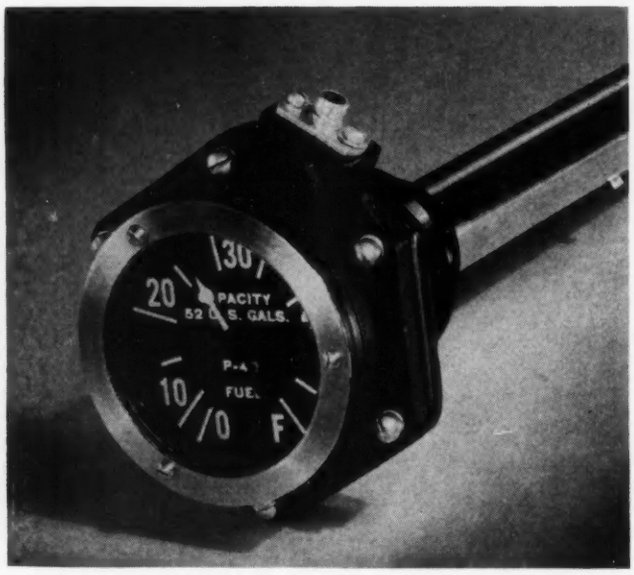
The remaining item of mold cost should be analyzed from the standpoint of cost per molded part. To do this it is important to have a true estimate of the total number of parts to be molded. The cheapest mold may wear out before the order is complete while the better built mold will have unnecessarily long life. A typical example is the customer who asks for the cheapest unhardened hand mold to make a hundred parts. After reordering a few times he suddenly complains about tolerances to the embarrassment of everybody.

A well planned mold has the proper number of cavities to give the required production capacity. Too few cavities may mean the otherwise unnecessary construction of an additional mold. Too many cavities are as bad as too few. The law of diminishing returns also places a physical limit on the ultimate number of cavities for one mold. Slower molding operation is unavoidable for multi-cavity molds and large presses. Not infrequently the same number of cavities divided between two frames will produce more parts at the same price per part. Multi-cavity molds are more susceptible to a damaged cavity which can cause complete production shut down.

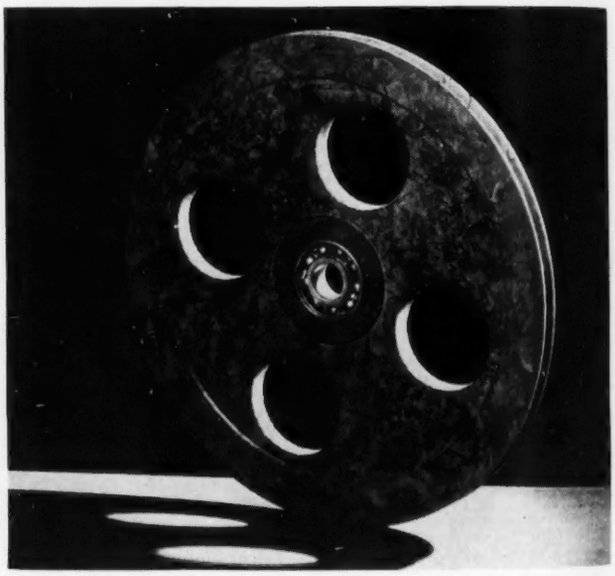
Barring abuse the life of a mold is limited by wear of the molding surfaces. Worn pins can be replaced quickly, but when a cavity becomes too large, an expensive replacement may follow. Hard chromium plating is a particularly useful tool for extending the life of mold cavities. A thickness of 0.001 in. on new molds lasts indefinitely. Worn surfaces of old molds can be rebuilt to size with a layer of chromium 0.010 in. or more thick.

Setting the Price

Having designed a part and selected a mold, a few words are in order on methods of price determination. Quotations on molded parts are divided into separate prices for parts and mold. Parts prices are built up from estimates of actual time required for each molding and finishing operation. Given the total time per molded part, rate of pay for labor, cost of material, overhead and profit, a straightforward price is obtained. In like manner mold costs are fixed by estimating hours and material for each individual part of the mold. By adding on assembly cost and overhead, total mold cost is determined. All prices are checked to be sure they are in line with similar jobs. Molds may range from one hundred to several thousand dollars; parts prices from a fraction of a cent to many dollars.

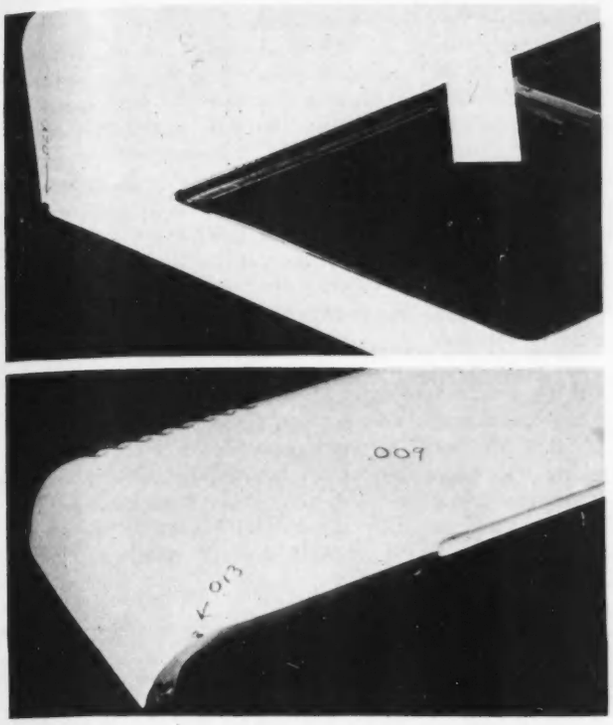


Important optical properties are transparency, high index of refraction, and double refraction. Typical application is this aircraft gasoline gage which has double acrylic windows; upper window can easily be replaced when it becomes scratched. Transparent plastics are used in liquid level gages, gas mask lenses, aircraft windows, meter windows, color filters. White urea compounds have optical applications in lamp diffusors. Other familiar optical uses are "piping" of light and models for photoelastic stress analysis.

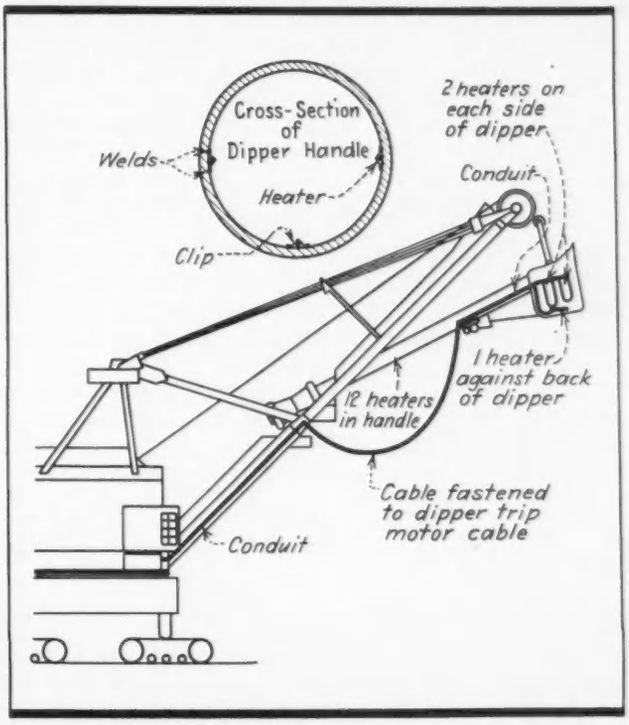


Thermal applications of plastics are limited by heat resistance, thermal conductivity, and thermal expansion. Outstanding example of heat-resistance is molded pulley over which hot coated wire passes. Thermoplastic resins are limited to maximum temperatures ranging from 140 to 250 deg. F., and thermosetting resins are good up to 170-350 deg. F., depending upon compounds. Asbestos or glass filled plastics and cold molded materials go higher. Low thermal conductivity is a detriment when heat must be dissipated.

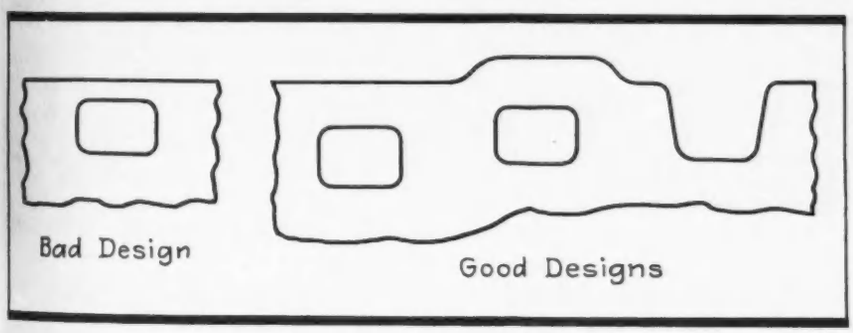
Causes and Cures



Excessively thick porcelain enamel coatings chip quite easily during assembly. Assembly losses are nearly seven times greater on parts with coatings over 0.021 in. thick than on parts with coatings less than 0.018 in. thick. Chipping of left-hand corner of part in top picture was caused by too thick a coating (0.028 in.). In part shown in lower picture, failure was not due to improper thickness (coating was 0.013 in. thick), but to severe assembly strains. Design details should limit chance of damage when drawing up bolts.



Cold weather damage to a big 30-yd. stripping shovel was caused when chilled metal of the dipper handle snapped during winter operations. Cure was to install twelve 2,000-watt, 230-volt heaters around inner surface of dipper handle. Lines of 4 heaters are placed in bottom and sides of handle. Heaters were also installed in dipper itself, reducing accumulation of frozen mud. Thermostat was set to close circuit at 25 deg. F. since no trouble was experienced above that temperature. Another set of heaters kept the cab warm.



Cored holes near edges of a plastic molded mechanical part caused trouble. Plastic would not flow as desired between the core and the wall of the mold. This is especially true with high-impact types of plastics, filled with cotton linters or shredded fabric. Such materials require high molding pressures and will not flow properly through narrow openings. If resin content only is squeezed into the narrow space, the thin section will be even weaker than if a wood-flour filled, general-purpose plastic were used. Above are three methods to avoid these too-thin sections. Choice of first or second method depends upon hole location. Third method is the simplest and may eliminate an expensive core.

RELIEF VALVE which adjusted the pressure in a standard filter press used for filtering clay and graphite mix wore out in a short time. Abrasive action of clay and graphite in suspension gullied conical plug and seat. Valves of various materials were tried. Steel rusted so badly it was dropped. Bronze and Monel were tried, but were unsatisfactory. Even sapphire valves wore out too soon. Finally a cure was found by adding a mechanism which rotated valve plug slightly with every pump stroke. This prevented gullying by distributing abrasive wear evenly. Moreover, rotation seemed to lap in the plug and seat. Bronze valves so rotated lasted one year instead of only a day.

PRODUCT ENGINEERING will pay a minimum of $3 for each example published in Causes and Cures. Where illustrations are necessary, include drawings, rough sketches or photographs.

HYDRAULIC CYLINDERS – II

JAMES E. THOMPSON and RODNEY B. CAMPBELL

PRELIMINARY design considerations were described in Part I of this article which appeared in March PRODUCT ENGINEERING. This second part of the article deals mainly with sealing methods for various types of joints.

End-Cap Sealing Methods

Choice of end-cap sealing method governs the design of the cylinder barrel of the end-cap to a considerable degree. In Fig. 12 are shown end-cap sealing methods and in Fig. 13 cylinder barrel designs.

Thickness of the cylinder wall is governed by the maximum working pressure encountered and a design pressure of 150 percent of the maximum working pressure should be used to provide for surge conditions. However, the wall thickness of a steel barrel should not be less than $\frac{1}{8}$ in. nor should it be less than $\frac{3}{32}$ in. for an aluminum alloy barrel, regardless of the minimum usable thickness shown by stress analysis. It is unreasonably expensive to machine cylinder barrels thinner than these values. The thickness of the barrel at the root diameter of threads should be 150 percent of the nominal wall thickness in order to compensate for the weakening effect of the threads.

The sealing method in Fig. 12(A) is the best arrangement because it provides a positive seal that can be adjusted without disturbing the cylinder heads. Essential elements of this seal are a stuffing box formed by cylinder end-cap, barrel and packing nut, with the packing being a triangular ring with a steel follower. When packing nut is tightened, the follower forces the packing tightly into gland to form a positive seal.

Important points to observe when designing this type of seal are: (1) the packing ring must be in the form of a truncated equilateral triangle, and the packing follower cross-section must be a right triangle having an angle of 40 deg. between base and hypotenuse. This arrangement gives a positive compression of the packing upon tightening the packing nut, with the flat-top on the packing providing considerable travel of the follower before it bottoms in the stuffing box.

The ring-seal shown in Fig. 12(B) is an excellent seal for end-caps which are semi-permanently installed, such as those for hydraulic shock-absorber cylinders. The seal shown in Fig. 12(C) is an original design of Rodney B. Campbell, and is excellent for large

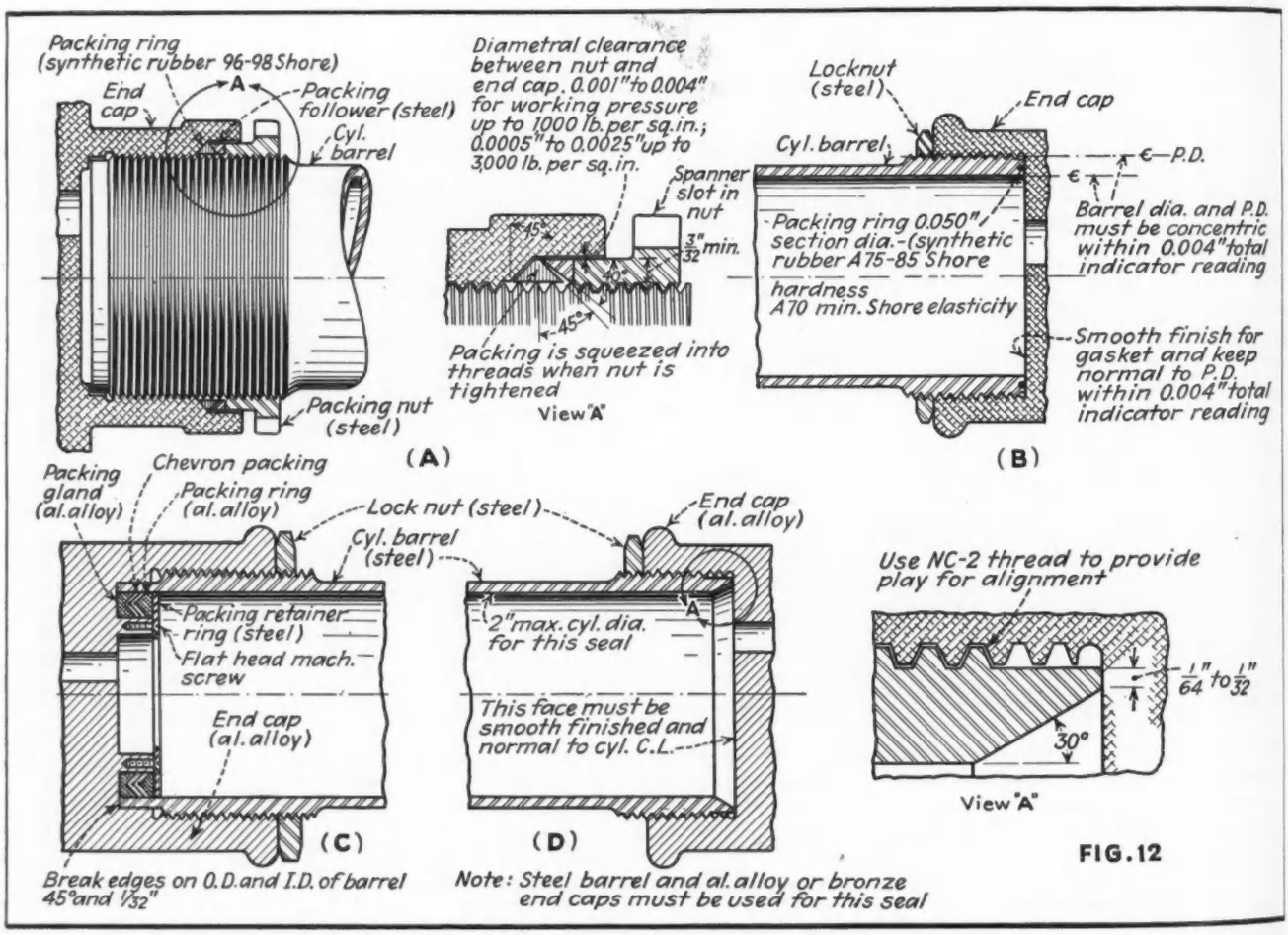

Packing ring (synthetic rubber 96-98 Shore)
End cap
A
Packing follower (steel)
Cyl. barrel
Packing nut (steel)
(A)
Diametral clearance between nut and end cap. 0.001" to 0.004" for working pressure up to 1000 lb. per sq.in.; 0.0005" to 0.0025" up to 3,000 lb. per sq.in.
Spanner slot in nut
45°
3/32" min.
45°
Packing is squeezed into threads when nut is tightened
View "A"
Locknut (steel)
End cap
Cyl. barrel
P.D.
Barrel dia. and P.D. must be concentric within 0.004" total indicator reading
Packing ring 0.050" section dia.-(synthetic rubber A 75-85 Shore hardness A 70 min. Shore elasticity
Smooth finish for gasket and keep normal to P.D. within 0.004" total indicator reading
(B)
Packing gland (al. alloy)
Chevron packing
Packing ring (al. alloy)
Lock nut (steel)
Cyl. barrel (steel)
End cap (al. alloy)
Packing retainer ring (steel)
Flat head mach. screw
End cap (al. alloy)
(C)
Break edges on O.D. and I.D. of barrel 45° and 1/32"
2" max. cyl. dia. for this seal
A
This face must be smooth finished and normal to cyl. C.L.
(D)
Note: Steel barrel and al. alloy or bronze end caps must be used for this seal
Use NC-2 thread to provide play for alignment
1/64" to 1/32"
30°
View "A"
FIG. 12

Fig. 12—(A) Stuffing box seal for end-cap. (B) Ring seal for end-cap. (C) Chevron seal for end-cap. (D) Metal-to-Metal seal

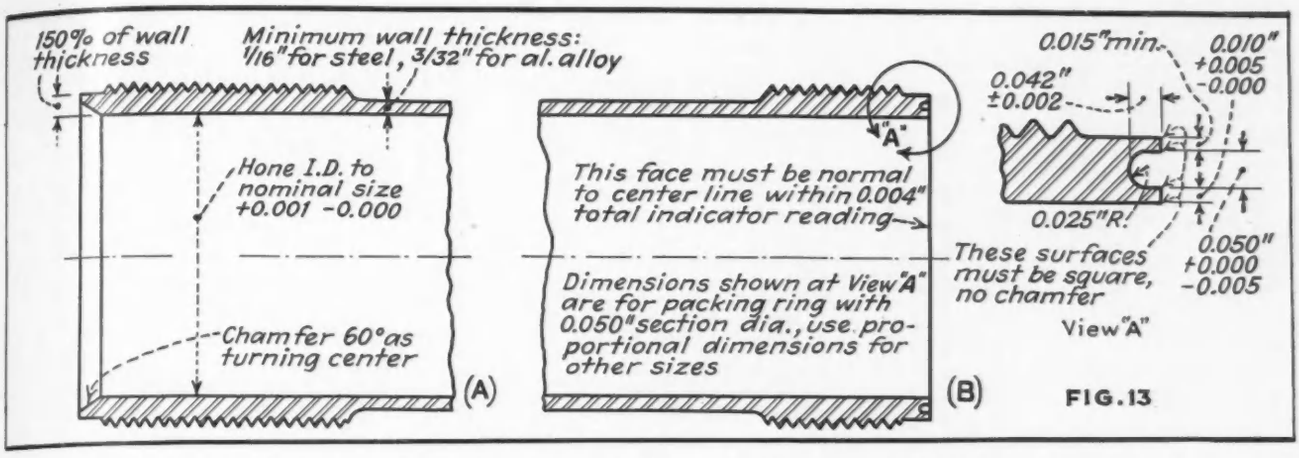

150% of wall thickness
Minimum wall thickness: 1/16" for steel, 3/32" for al. alloy
Hone I.D. to nominal size +0.001 -0.000
Chamfer 60° as turning center
(A)
"A"
This face must be normal to center line within 0.004" total indicator reading
Dimensions shown at View "A" are for packing ring with 0.050" section dia., use proportional dimensions for other sizes
(B)
0.015" min.
0.010" +0.005 -0.000
0.042" ±0.002
0.025" R.
0.050" +0.000 -0.005
These surfaces must be square, no chamfer
View "A"
FIG. 13

Fig. 13—(A) Design of cylinder barrel for stuffing box seal. (B) Design of cylinder barrel for ring seal

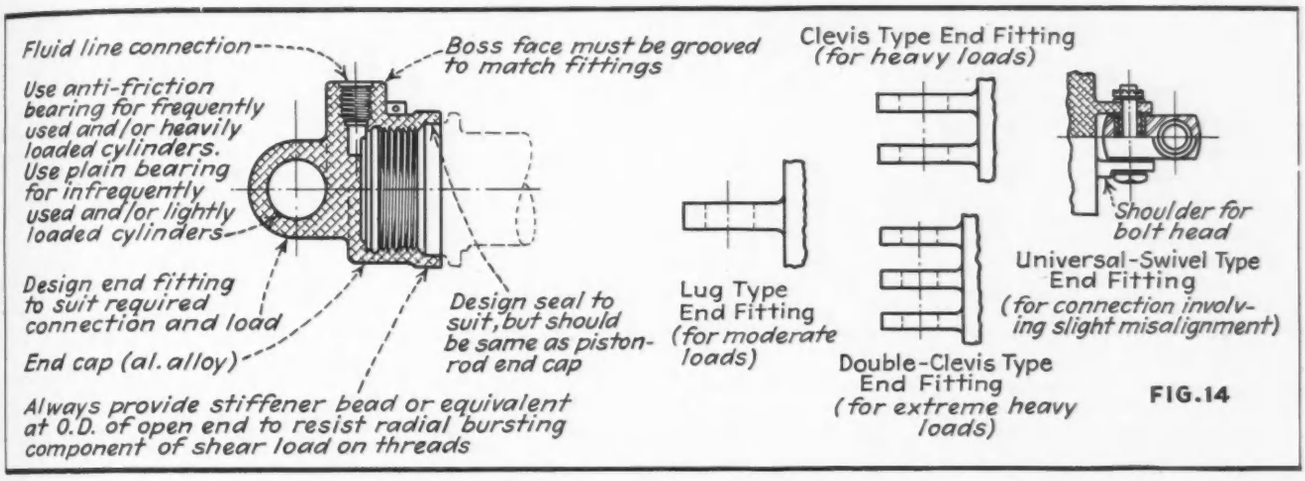

Fluid line connection
Boss face must be grooved to match fittings
Use anti-friction bearing for frequently used and/or heavily loaded cylinders. Use plain bearing for infrequently used and/or lightly loaded cylinders
Design end fitting to suit required connection and load
End cap (al. alloy)
Design seal to suit, but should be same as piston-rod end cap
Always provide stiffener bead or equivalent at O.D. of open end to resist radial bursting component of shear load on threads
Clevis Type End Fitting (for heavy loads)
Lug Type End Fitting (for moderate loads)
Shoulder for bolt head
Universal-Swivel Type End Fitting (for connection involving slight misalignment)
Double-Clevis Type End Fitting (for extreme heavy loads)
FIG. 14

Fig. 14—Cylinder head end-cap details showing proportions of essential elements

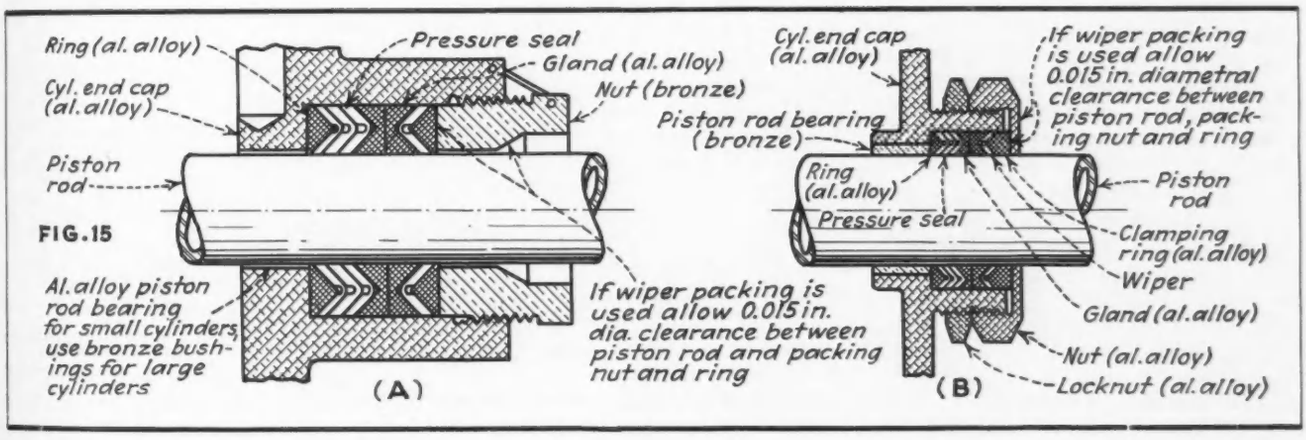

Ring (al. alloy)
Pressure seal
Gland (al. alloy)
Nut (bronze)
Cyl. end cap (al. alloy)
Piston rod
FIG. 15
Al. alloy piston rod bearing for small cylinders, use bronze bushings for large cylinders
(A)
If wiper packing is used allow 0.015 in. dia. clearance between piston rod and packing nut and ring
Cyl. end cap (al. alloy)
Piston rod bearing (bronze)
If wiper packing is used allow 0.015 in. diametral clearance between piston rod, packing nut and ring
Ring (al. alloy)
Pressure seal
Piston rod
Clamping ring (al. alloy)
Wiper
Gland (al. alloy)
Nut (al. alloy)
Locknut (al. alloy)
(B)

Fig. 15—(A) Plug-type piston rod packing gland for use whenever space limitations permit. (B) Cap-type piston rod packing gland

bore cylinders. The metal-to-metal seal shown in Fig. 12(D) is suitable for cylinders having a maximum bore of 2 in. and working pressures up to 1,000 lb. per sq. in. The factors influencing the success of this seal are: (1) a narrow knife-like edge on the end of the barrel, to cut into the end cap upon tightening; (2) a sufficiently loose thread between cap and barrel to provide for slight misalignment of barrel-end and end-cap face.

End-Cap Design

Essential elements of a suitable end-cap design are shown in Fig. 14, while details of packing glands for the piston-rod end-cap are shown in Fig. 15(A) and (B). The plug-type packing gland shown in Fig. 15(A) is the less expensive to manufacture and should be used whenever space limitations permit. When space limitations are critical the cap-type packing gland shown in Fig. 15(B) can often be used to advantage. In either arrangement it is important to

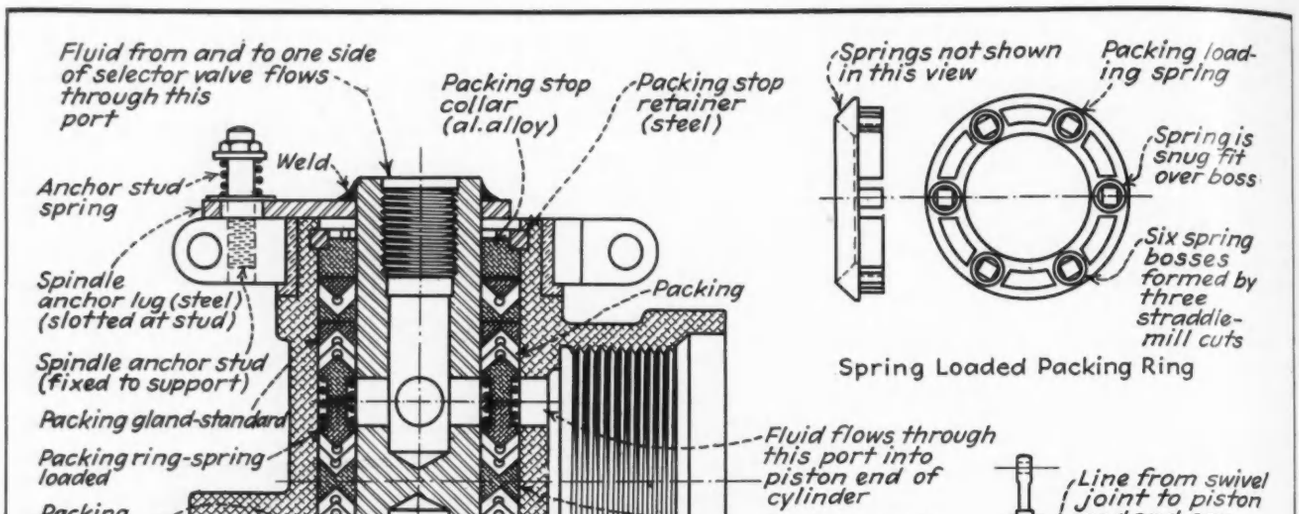

Fluid from and to one side of selector valve flows through this port
Packing stop collar (al.alloy)
Packing stop retainer (steel)
Weld
Anchor stud spring
Spindle anchor lug (steel) (slotted at stud)
Spindle anchor stud (fixed to support)
Packing gland-standard
Packing ring-spring loaded
Packing loading spring
Rigid line leads from this port to piston rod end of cylinder
Packing
Fluid flows through this port into piston end of cylinder
Packing gland
Design end cap seal to suit, but should be same for both caps
Cylinder support (steel) design to suit structure
Support bearing (bronze)
Packing retainer collar (steel)
Spindle (steel)
Packing ring-standard
Retainer collar locknut (steel)
Fluid from and to one side of selector valve flows through this port
FIG.16
Springs not shown in this view
Packing loading spring
Spring is snug fit over boss
Six spring bosses formed by three straddle-mill cuts
Spring Loaded Packing Ring
Line from swivel joint to piston rod end cap (rigid)
Lines to selector valve (rigid)
Diagram of Actuating Cylinder with Swivel Joint
Note: Diametral clearances same as those given elsewhere for piston packing gland ports

Fig. 16—Details of swivel joint that is incorporated within the end-cap of actuating cylinders

provide an adequate bearing for the piston rod. When the cylinders are of small bore the aluminum cylinder head wall is often sufficient, but a bronze bushing must always be used for large-bore, heavy-duty cylinders. The omission of a wiper packing has previously been discussed and is governed entirely by the conditions under which the cylinder will operate.

End-caps always contain the bosses and ports for connection of the fluid lines. If possible, the boss wall thickness should be sufficient to permit tapping out for the next larger size pipe or fitting. This will avoid the necessity of discarding an end-cap in the event of the threads being stripped. If the cylinder is designed for aircraft use, the ports should be machined with a suitable straight-thread and grooved-face for AN standard fittings, or with pipe threads if the cylinder is intended for industrial or automotive use.

Swivel Joints

A fluid swivel joint incorporated within the end-cap of a hydraulic actuating cylinder is shown in Fig. 16. This swivel joint provides for conducting the hydraulic fluid into the cylinder without the necessity of using high-pressure hose assemblies between the cylinder and the rigid plumbing lines. The use of swivel joints is not common for small actuating cylinders but with large actuating cylinders the flexible lines become quite unwieldly and a swivel joint is a decided advantage in many respects.

The swivel joint is installed concentric with the pivot-axis at the cylinder's fixed end, with the hydraulic pressure lines connecting to each end of the stationary spindle. Passageways within the joint conduct fluid directly to the piston head, while a short, rigid line extends the length of the cylinder barrel to conduct fluid from the swivel joint to a port in the piston rod end-cap.

In Fig. 16 is shown a swivel joint consisting of three principal parts: (1) the housing, formed by the end cap; (2) the stationary spindle, fixed to one of the cylinder supports by means of the spindle anchor-lug; (3) the packing group that permits fluid transfer and rotation between spindle and housing.

The housing bore is ported to conduct fluid into the actuating cylinder and is ground to an exact diameter to receive the packing group. The packing group comprises packing, spacers, glands, and springs, and provides two sealed circumferential channels extending entirely around the inner surface of the housing bore. The fluid ports register with these channels. The packing loading-springs insure positive sealing at low fluid pressures. The spindle passes through the center of the packing group, is located within

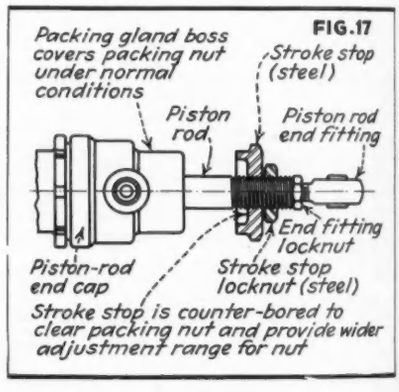


FIG.17
Packing gland boss covers packing nut under normal conditions
Stroke stop (steel)
Piston rod
Piston rod end fitting
End fitting locknut
Piston-rod end cap
Stroke stop locknut (steel)
Stroke stop is counter-bored to clear packing nut and provide wider adjustment range for nut

Fig. 17—Stroke-stop consisting of an adjustable collar threaded to the piston rod

the housing by the spindle anchor lug, and bears against the packing rings and glands. The packing retainer collar is machined to an I.D. sufficiently large to clear the spindle and thus avoid the spindle bearing against the collar with resultant working of the threads.

It is sometimes desirable to provide stroke adjustment for an actuating cylinder. This can be accomplished without difficulty on the retracting stroke by a stroke-stop collar, as shown in Fig. 17. The provision of an integral, adjustable stroke-stop for the extension stroke is no easy matter and should be incorporated in the mechanism being operated. The stroke-stop shown in Fig. 17 consists of an adjustable collar threaded to the piston rod and a heavy boss forming part of the end-cap. This boss completely incloses the packing gland nut and provides a shoulder for bottoming the stroke-stop collar. The collar should be counterbored to provide additional clearance for the packing gland nut.

Dash-Pot Design

Ordinarily a dash-pot will be required on only one end of a cylinder. although the piston shown in Fig. 18 is provided with dash-pots at both ends to cover all conditions. The elements of dash-pot design are obvious upon inspection of Fig. 18, and simply involve trapping a portion of the fluid near the affected end of the piston's stroke, to be forced through a suitable orifice in the same manner as a hydraulic shock absorber strut. The piston movement is unrestrained until the end of the dash-pot strikes against the actuating cylinder's piston head, and it then becomes necessary to expel the fluid behind the dash-pot piston through the orifice provided, which acts as an energy absorber. The orifice for the piston-rod end dash-pot shown in an annulus between the dash-pot piston and the cylinder wall, and clearance between the parts must be small, in the order of 0.010 in. max., if adequate energy absorption is to be expected.

Ring Seal Designs

The information in the preceding paragraphs was based upon the conventional "chevron-seal" form of actuating cylinder design and only in the piston design shown in Fig. 9(E) has the ring-seal design been considered. This form of seal has shown promising results and its further development may lead to actuating cylinder designs permitting greatly reduced manufacturing cost and much smaller cylinder dimensions.

One manufacturer of hydraulic actuating cylinders is producing cylinders using a single sealing ring for the piston with another single-ring seal for the piston rod. This arrangement eliminates the necessity of packing adjustment nuts and thereby reduces the overall length of a given cylinder, as compared with a conventional chevron-seal design. Also, the manufacturing cost of the ring-seal cylinder would be less.

This cylinder design form, however, is still in the experimental stage, hence its use should be approached with caution and considered only after successfully completing extensive life tests of each new design. Also, the patent situation regarding this form of seal appears rather involved at present, and should be thoroughly investigated before planning its use.

Clearances for Chevron Packing Parts

All clearances listed below are radial, not diametral, and are for working pressures up to 1,500 lb. per sq. in. Use 50 percent of these values for working pressures between 1,500 and 3,000 lb. per sq. inch.

To find the recommended clearance between two members read across from one to the column containing the other. All values are in thousands of an inch, thus 2—4 signifies 0.002 to 0.004 inches.

PISTON PACKING

	Piston Body	Cylinder Wall	Packing Gland	Packing Ring
Piston body	....	2– 4	2–6	4–10
Cylinder wall	2– 4	....	2–7	4–11
Packing gland	2– 6	2– 7	....	....
Packing ring	4–10	4–11	....	....

PISTON ROD PACKING

	Piston Rod	Stuffing Box Bore	Packing Gland	Packing Ring
Piston rod	....	....	2–7	4–11
Stuffing box bore	....	....	2–6	4–10
Packing gland	2– 7	2– 6	....	....
Packing ring	4–11	4–10	....	....

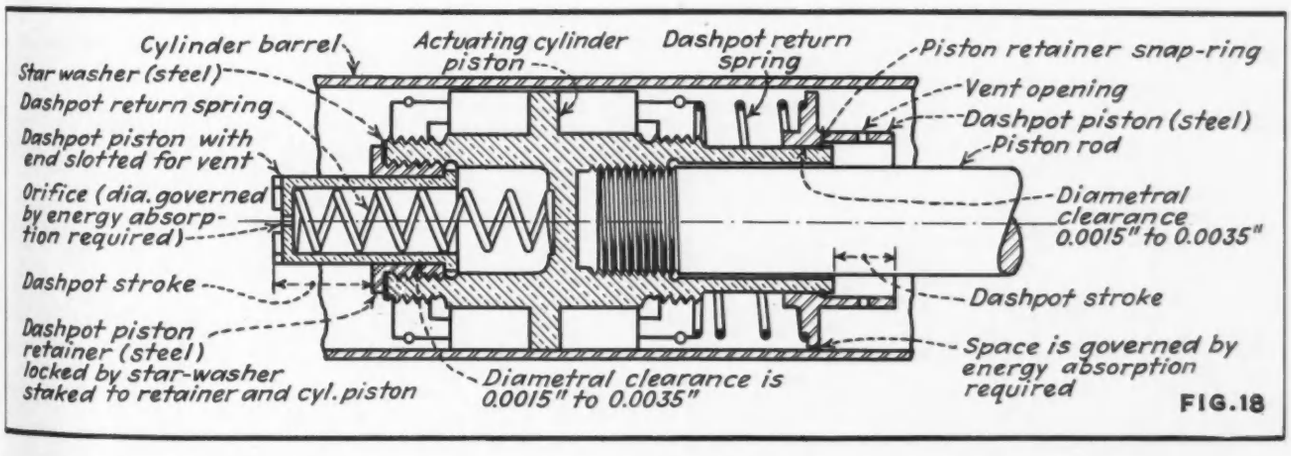

Cylinder barrel
Actuating cylinder piston
Dashpot return spring
Piston retainer snap-ring
Star washer (steel)
Dashpot return spring
Vent opening
Dashpot piston (steel)
Dashpot piston with end slotted for vent
Piston rod
Orifice (dia. governed by energy absorption required)
Diametral clearance 0.0015" to 0.0035"
Dashpot stroke
Dashpot stroke
Dashpot piston retainer (steel) locked by star-washer staked to retainer and cyl. piston
Space is governed by energy absorption required
Diametral clearance is 0.0015" to 0.0035"
FIG. 18

Fig. 18—Dash-pot design for trapping a portion of fluid at end of piston stroke to serve as a shock absorber

FITS AND TOLERANCES—

DR. ERNESTO GEIGER
Consulting Engineer

SOME EXAMPLES of current practice in specification of fits and tolerances. Both the American tentative standard, referred to as ATS, and the International Standards Association system, known as ISA, are indicated. Although only ATS class numbers are shown on diagram, the ISA designations in the following table were originally specified. Common applications for each example of fit are also tabulated. See also page 305 for charts to determine tolerances and allowances for different classes of fits for various nominal dimensions required in engineering practice.

Classes of Fits, Designations and Applications

ATS CLASS [1]	NAME	ISA DESIGNATION [2]	APPLICATIONS OF FITS ILLUSTRATED
8	Heavy force fit	H7r6	Wrist pins, armature shafts in electric motors, shaft couplings and flanges (Figs. 1, 14).
7	Medium force fit	H7n6	Bushings, friction clutch disks, engine cranks, cylinder liners, eccentrics on their axles, and wheels of locomotives and cars (Figs. 2, 3).
6	Tight fit	H7m6	Keyed gear hubs, axially stressed bushings, jig bushings, pulleys and rocker arms (Figs. 3, 12).
5	Wringing fit	H7k6	Keyed hubs of gears and pulleys, keyed shaft couplings, handwheels and cranks, locating keys, tongues and dowels (Figs. 4, 9, 10, 19).
5–4	Intermediate	H7j6	Centrifugal rotors, pulley hubs, centering pins and rims, and renewable bushings and collars (Figs. 4, 6, 11).
4	Snug fit	H7h6	Automotive shift gear hubs, milling cutters on mandrels, centering pins and rims for frequently dismantled machines, centrifugal pump rotors, interchangeable bushings, bearing caps, shims and body and jaws of drill chuck (Figs. 5, 7, 8, 10).
4_1	Large tolerance	H8h8	Agricultural and mining machinery, interchangeable parts not rotating on shafts, stop collars for shafts, hand cranks and heavy shaft couplings (Figs. 15, 16, 19).
4_2	Extreme tolerance	H11h11	Heavy machinery such as hoists, cranes, etc. (Figs. 14, 19).
4–3	Intermediate	H7g6	Cross heads in their slides, sliding gears, valve plungers and valve lifters, and eccentrics in their collars. (Figs. 2, 10).
3	Medium fit	H7f7	Shafts rotating at less than 600 r.p.m. with bearing loads under 600 lb. per sq. in. (Figs. 2, 3, 6, 10).
3_1	Large tolerance	H9f8	Bushings for vehicle wheels, heavy bearings, hydraulic pistons and bearing width (Figs. 15, 19).
2	Free fit	H8e7	Bushings on multiple-supported transmission line shafts, crankshaft bearings, oscillating bearings and quick acting slides (Figs. 7, 10).
2_1	Large tolerance	H10e9	Material handling machinery and stamped pieces from metal sheets. Used also on electric switches (Figs. 13, 14).
1	Loose fit	H8d10	Idle pulley bushing quick acting plungers, valve lifters and turbo-generator axle-bearings (Figs. 5, 12, 18).
1_1	Large tolerance	H11d11	Hoists and other material handling machinery (Fig. 17).

[1] See ATS B4a — 1925, "Tolerances, Allowances and Gages for Metal Fits," American Society of Mechanical Engineers and American Standards Association, 29 W. 39th St., New York; price 50 cents.

[2] See "Tolerances for Cylindrical Fits" by John Gaillard *Industrial Standardization*, January to April, 1941.

Examples of All Classes

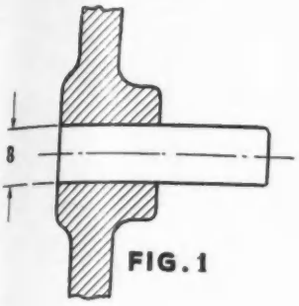


FIG. 1

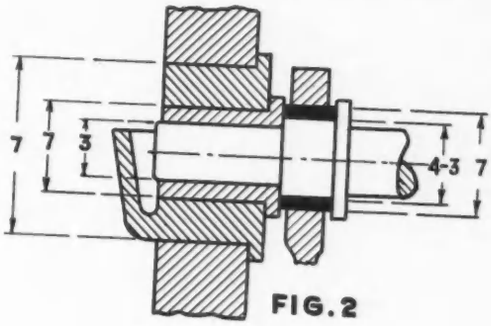


FIG. 2

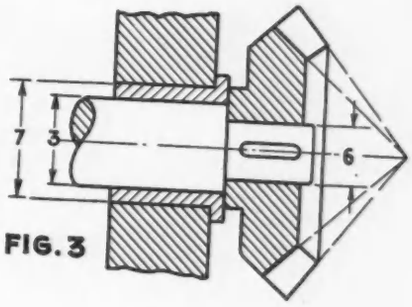


FIG. 3

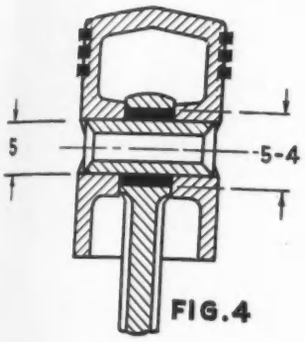


FIG. 4

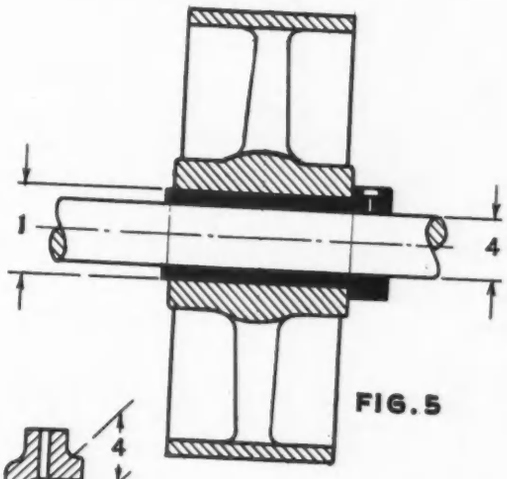


FIG. 5

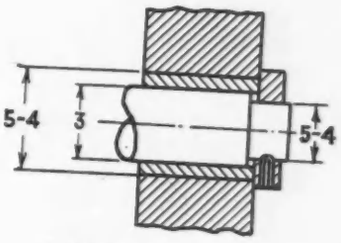


FIG. 6

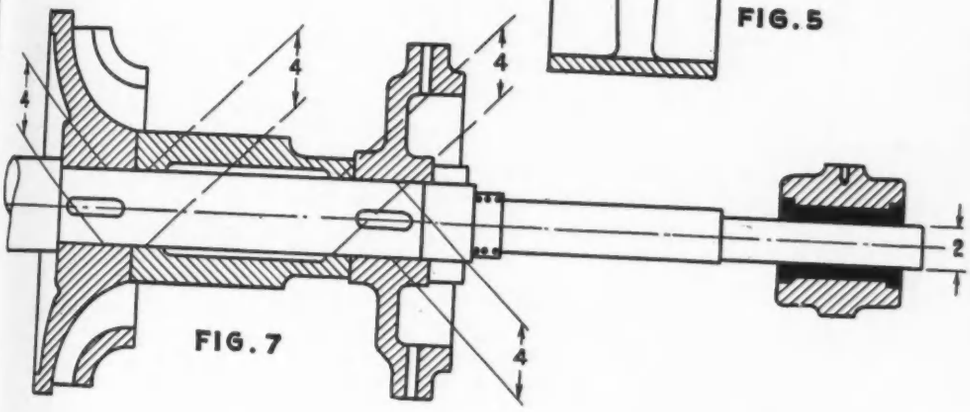


FIG. 7

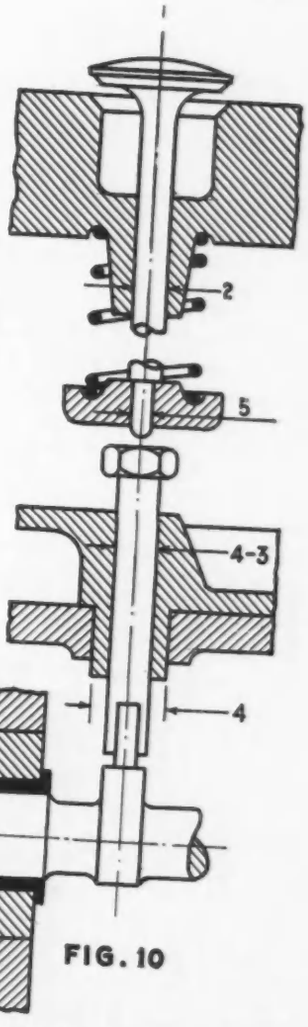


FIG. 10

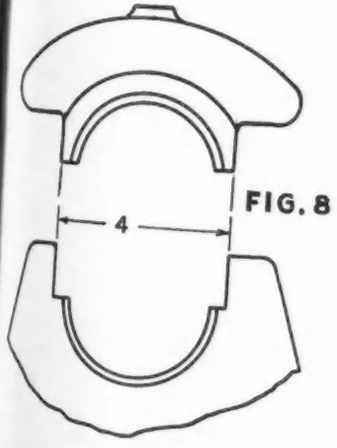


FIG. 8

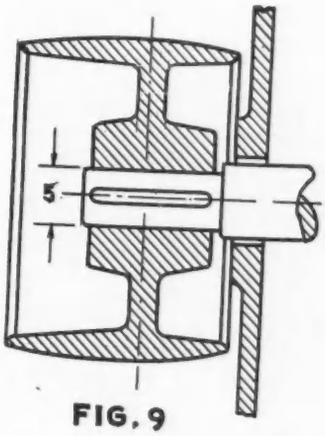


FIG. 9

(*Continued on next page*)

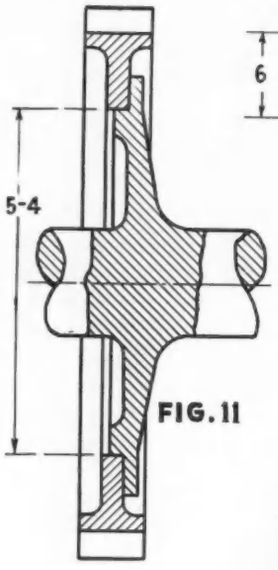


FIG. 11

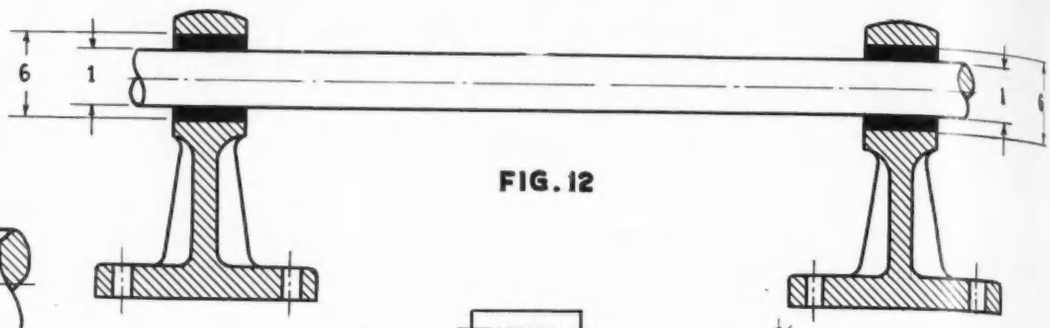


FIG. 12

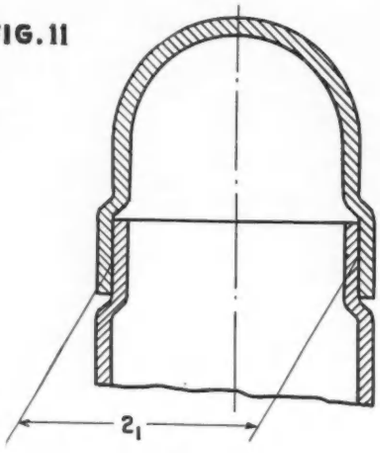


FIG. 13

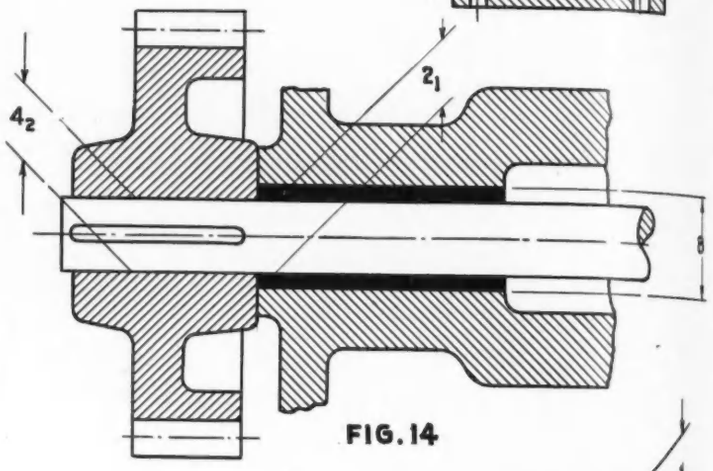


FIG. 14

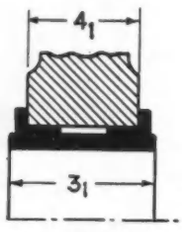


FIG. 15

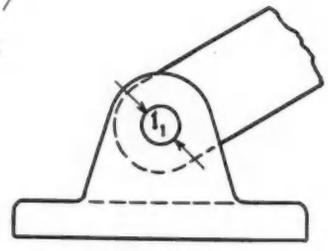


FIG. 17

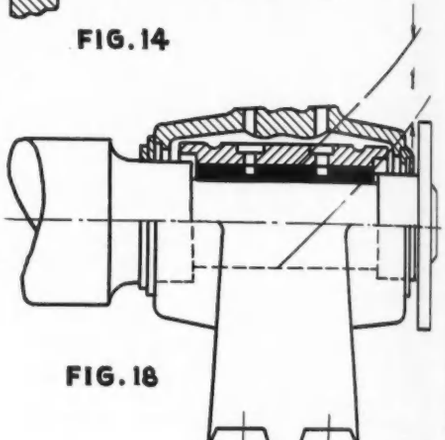


FIG. 18

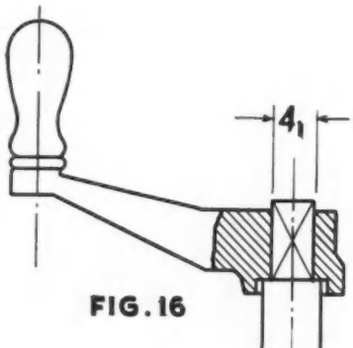


FIG. 16

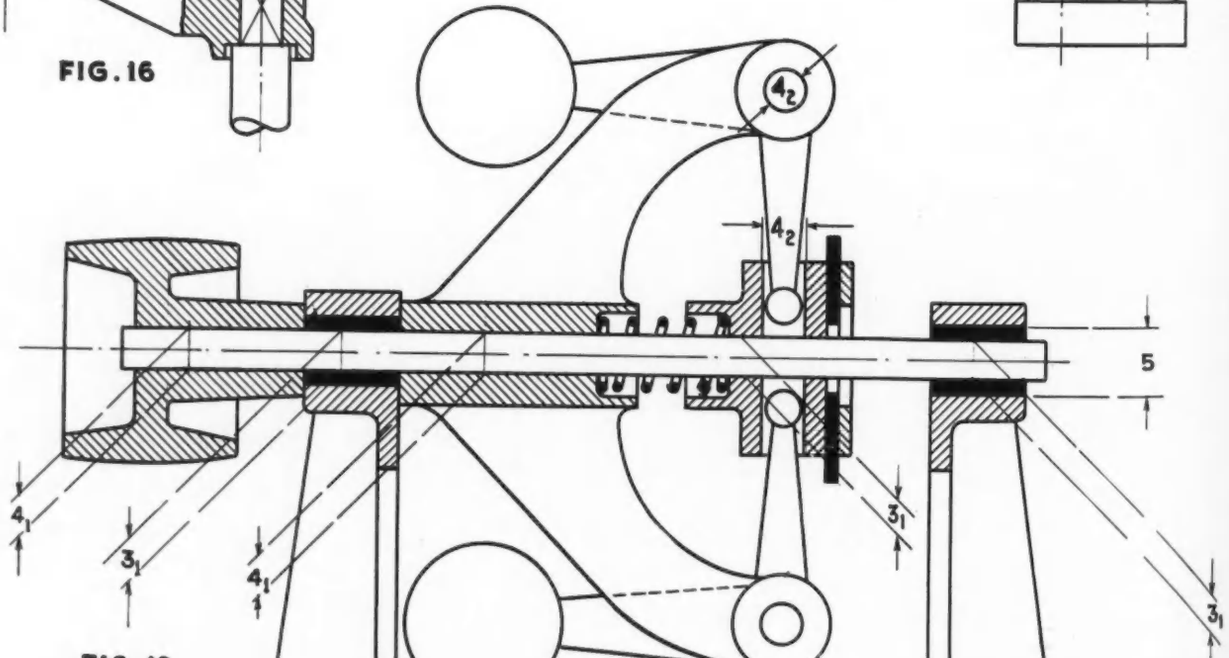


FIG. 19

PATENT PROCEDURE – IV

Guarding Your Rights While Making An Invention

MILES HENNINGER
Patent Department, Allis-Chalmers Manufacturing Company

NOW that the preliminary investigation or so-called "pre-ex" has failed to show that some other person solved your problem in the same way you did and before you did, or, in patent office parlance, that your invention is anticipated by the prior art, you are ready to complete your invention. But, you say, my invention is complete and ready for manufacture and I have done everything that I can. Consider the things you have so far done. You have:

(a) recognized a problem or a desirable result to be achieved;

(b) conceived a mental picture of how to attain your result;

(c) made sketches and have written a description of your mental concept or picture;

(d) shown your sketches to other persons and had them read your description and had them witness your signature; that is, you have disclosed your invention to others;

(e) made a search and failed to find prior devices achieving your result in the same way you did, and hence you believe your invention to be new.

Acts (a) and (b) are only your mental processes and count for nothing if you stop there. In act (c) you have made a physical record of the results of your cerebrations and by act (d) you are able to prove the date and subject matter of your mental concept. And act (e) is useful only for your own information and, if made by another person, will furnish additional proof that you disclosed your ideas to such other person. You have a problem and plan for its solution, but you have not yet actually solved the problem. You have provable mental concepts, but you do not have embodiment of your concepts in physical materials such as metal, wood or other materials. You have conception of the invention, but the invention is not yet born—it is not yet reduced to practice. The first half of the process of making an invention is finished, but the last half must now be done before your invention is complete.

Reduction to practice of your invention must now follow before your invention is made. That reduction to practice may be construction and operation of a working model (or so-called pilot machine or plant regardless of size) or a full size structure which will practically demonstrate that your art (process) works, that your machine produces the desired result, that your manufacture is what you thought it would be or that your composition has the qualities you imagined for it. The building of a working model or full size structure is actually reducing the invention to practice and is desirable when the invention is complex or involves heretofore untested principles of design. However, under some circumstances, it is not desired to build an actual model or structure, and some other act must be done to reduce the invention to practice; i.e., to complete the invention. The only other act recognized by the Patent Office as a reduction to practice is the filing of a patent application which the Patent Office construes or considers as a reduction to practice. It is therefore necessary that the inventor now determine whether he wishes to reduce his invention to practice by:

(a) building a working model (pilot machine or plant) or full size structure; or

(b) filing a patent application thereon.

If the inventor decides to build a working model or full size structure, the work should be commenced immediately after the preliminary search has been made and should be continued to completion as diligently as is humanly possible. The inventor may avail himself of any assistance within reach or may do all the work himself but he must himself keep proper records of the progress of the work. Let us assume that the inventor is not a draftsman, does not have mechanical ability and wishes to utilize the abilities and facilities of others. The first step will probably be the preparation of working drawings to scale and in such detail as are necessary for another person to work from. The inventor's sketches and his description already made for the purpose of disclosing the invention to others and making the preliminary search are the basis for the draftsman's work. If, during preparation of the drawings, the draftsman or any other person suggests changes in details of design, the inventor may accept any such suggestions as are within the ordinary and usual skill of the draftsman or designer and such changes to improve the working of the device are the property of the inventor.

The fears of some inventors that their inventions will be stolen by men working on them are baseless if proper records are kept. To keep a record, the draftsman should place on each sheet of the drawings a statement that:

I, *John Jones*, certify that I was employed by *Samuel Smith*, residing at *1014 N. 126th Street, Milwaukee, Wis.*, on *February 29, 1942*, to make working drawings for the completion of an invention relating to *internal combustion engines*, that the said invention was fully explained to me by sketches and a written description, that I disclaim being inventor or coinventor of any part of said invention, and that I completed each sheet of drawings relating to said invention on the date following my signature hereto.

John Jones
Draftsman's Name

3/30/42
Date

The sheets of drawings should be correlated by marking, "6 sheets—sheet 1." The drawings should be completed by the draftsman with as little change in principle as possible from the original sketches. Duplicate prints or photostats of the drawings should be made for use in further development but the original drawings should be signed and dated by the inventor and filed in a sealed envelope in a secure place. If any major changes are to be made from the original set of drawings, a complete new set of drawings should be made and they should be certified, marked and otherwise handled as in the case of the first set of drawings.

Write Agreements

Now comes the process of translating the working drawings into the actual physical structure. The patterns for the parts to be cast, the casting of the parts and the machining and final assembly of parts to make the completed struc-

ture may be done by any one whom the inventor may select. But it is desirable for the inventor to make a written agreement with the pattern maker, founder and machinist that all improvements made during construction shall be the property of the inventor. Such agreement is essential when a separately patentable improvement results. It is necessary that a record be kept of the work done and the dates relating to such work. Orders given by the inventor for work to be done should be written and acceptances of such orders should also be written. All writings relating to the invention should be signed and dated by the person making the writing. The procurement of materials should be by written order which should relate only to the invention. Payments for materials or work should be receipted in writing. Generally, there should be a written record of all transactions relating to building of the model or full size structure and the inventor should keep originals or copies of all papers relating to such transactions.

Keep A Work Record

During building of the model or full size structure, the inventor should keep a daily diary or journal relating to the invention. The diary should be a true journal or daybook; that is, entries should be made regularly and even daily and should be a complete note of work done or that no work was done and the reason why nothing was done. The diary should be in ink and no erasures should be made in it, or for that matter, in any of the papers, especially after signing and witnessing. The book used should be permanently bound and not loose leaf, unless the inventor is willing to have his assistants initial and date each page. Even when permanently bound, the assistants should initial and date each page recording a change in the theory of or the structure showing the invention. A useful record is a series of photographs showing the various stages in building the invention, together with the inventor and the assistants then working on the invention. If the photographs are taken by a professional photographer, he should date and otherwise identify the photographs. If taken by another person, prints of the photograph should be signed and dated by all persons shown therein.

After the model or full scale structure has been made and tested, it should be carefully preserved as a whole and any persons witnessing the tests should evidence such witnessing by signing and dating the test record. Even though the first structure is not satisfactory, it should never be robbed of any of its parts, but should be kept intact. A partially complete model or machine has very little value as evidence regardless of the reason for its incompleteness when presented.

The whole object in keeping records is to provide incontrovertible proof, at some future date, that the drawings, description and model existed on the dates shown by the records in exactly the condition evident from the records. The inventor should carefully preserve all originals forming part of the record and should keep the dates of the various parts of the record (and of the patent application) secret as long and as much as possible.

After the working model or full scale structure is complete, the inventor may use any reasonable time for testing and may later sell it or otherwise put it into public use. A careful record of the date of sale or other public use must be kept because the patent application must be filed within one year after the date on which the specimen of the invention was sold (delivered and accepted) or otherwise put into public use. Just what constitutes a public use depends on the circumstances of each case. Any use of the invention, experimental or for convenience, pleasure or profit, becomes a public use when extended, either in time or number of instances, beyond what is reasonably necessary to demonstrate the practicability or usefulness of the invention. A single specimen or use by a single person may be sufficient to bar obtaining a patent. As in many other phases of patent law, it is best to define what is experimental use and therefore not public use. Generally, experimental use is any use in good faith to test the invention and it is immaterial whether or not the use is publicly visible (1) or only incidentally profitable to the inventor (2). The object and purpose of the inventor in placing the invention on sale or in use is the governing consideration as to whether or not the sale or use is public and not experimental.

Case Histories

In (1) Nicholson laid a strip of wooden pavement, in 1848, at a gate on a toll highway where heavy teams frequently stopped and started. He inspected the pavement and asked opinions of teamsters nearly every day for six years before filing his patent application. His use was held experimental and his patent valid.

In (2) Sprague devised a machine for making "arctic" overshoe buckles late in 1873 and used the machine in secret until 1878. The product of the machine, though imperfect, was sold and an application was filed in 1879, only after the machine was perfected. The machine was operated for profit and not only to test and improve it.

It is public use when the invention is used:

(3) in private by a person other than the inventor;

(4) by but not visible to the public;

(5) but not recognizable by the public;

(6) beyond a year's time without change;

(7) and unrestrictedly sold.

In (3) Barnes in 1855 gave his future wife a double leaf corset spring to use. He apparently felt no need to request that she keep the use secret. But the structure became generally used before the application was filed in 1866. Because of the single use which became public, the patent was held invalid.

In (4) Hall made and sold safes in 1860 using tapered bolts to fasten together the various plates, for which he filed an application in 1867. The court held the use to be public even though the bolts were concealed when in use.

In (5) a particular accelerator was used in vulcanizing rubber. The accelerator was destroyed in vulcanization of the tires and could not be identified from the finished tire as to composition or process of manufacture. More than two years (now one year) after the first use an application was filed. The use was held to be public.

In (6) Root invented a cable railway in 1876 in which the structure was buried in the ground and in use in 1878. The application was filed more than two years after 1878. No changes were made in the structure at any time, but in 1892 the inventor argued that the installation was still experimental and not publicly used. He lost.

In case (7), Ericsson invented a hot air engine. At least one engine was sold in 1877 for pumping water into the attic tanks of a residence, a job formerly left to the family coachman. The engine was frequently inspected and repaired free of cost to the owner. There was no guaranty of efficiency and no agreement to take it back if unsatisfactory and, after seven years, it was traded in on a new engine. The sale was without restriction and invalidated the patent on an application filed in 1880.

Footnotes

(1) Elizabeth v. American Nicholson Pavement Co. 97 U.S. 126

(2) Smith and Griggs Mfg. Co. v. Sprague 123 U.S. 249

(3) Egbert v. Lippmann 104 U.S. 333

(4) Hall v. Macneale 107 U.S. 367

(5) Grasselli Chemical Co. v. Nat. Aniline & Chemical Co. 29 F(2d) 305

(6) Root v. Third Ave. R.R. Co. 146 U.S. 210

(7) Delemater v. Heath 58 F 414

Materials Section

SILVER BRAZING ALLOYS and SOLDERS

HERBERT CHASE

Wider use of silver-bearing solders and brazing alloys is likely to result from restrictions on tin. Silver has long been used as an ingredient of brazing alloys because of the low melting point of these alloys and for such other advantages as good malleability and ductility, ability to make joints of great strength, good corrosion resistance and high electrical and thermal conductivity.

BRAZING ALLOYS

Of the brazing alloys, often called "hard silver solders," those that melt at 1,175 to 1,200 deg. F. and flow freely at or below 1,300 deg. F. are easiest to work with and possess most of the advantages of the silver brazing alloy series. Others have melting points up to 1,600 deg. F.

Low Melting Point not only decreases the time, heat and fuel required to make joints, but affords a larger factor of safety against over-heating the metals to be joined. The latter is of importance when brazing many non-ferrous metals, especially those based on copper, and particularly when in the form of thin sheets and tubes.

Malleability and Ductility are important not only in the joint itself, but also in permitting the brazing alloy to be rolled into thin sheets and strips or drawn into fine wire, thus contributing to convenience and economy in making joints. Since joints have maximum strength when the solder thickness in the joints is 0.001 to 0.003 in., as indicated in Fig. 1, little brazing alloy is needed.

Penetration. Silver imparts a penetrating quality to the alloy which is valuable for reaching points of minimum clearance. V-shaped clearances and fillets are not needed and only waste brazing.

Joints. Butt, lap and scarf joints are readily made. Many joints are made conveniently and economically by preplacing in the joint, or adjacent to it, a strip of wire of brazing alloy. When the joint is fluxed and heated the alloy penetrates the joint, flowing freely when the proper temperature is reached. Flux requirements are small.

Tensile Strength. Silver brazing alloys usually have a tensile strength of 40,000 to 60,000 lb. per sq. in. The significant factor, however, is the strength of the joint rather than of the alloy. In one set of tests all butt and lap joints, made with silver brazing alloy on Everdur (silicon-bronze) strip, had tensile strengths exceeding 56,000 lb. per sq. in. and broke outside, not at the joint. Tensile strengths of 80,000 lb. per sq. in. have been attained in test joints made on stainless steel.

Effect of Temperature. The tests described in the preceding paragraph were made at room temperature. Strength decreased in inverse proportion to temperature up to 700 deg. F., at which point it was 30,000 lb. per sq. in. According to one authority, 450 deg. F. is the maximum temperature

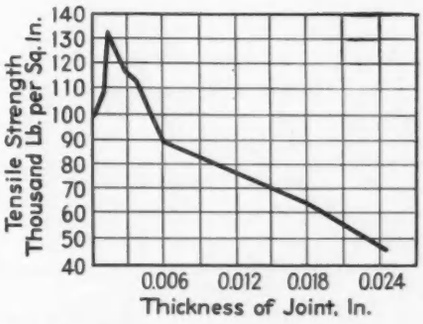


Fig. 1—Relation of joint thickness to tensile strength in butt joints of stainless steel to stainless steel

to which joints made with silver brazing alloy should be subjected on high pressure steam lines and still allow a reasonable factor of safety.

At least one boiler manufacturer uses a silver brazing alloy that melts at 1,175 deg. F. to make joints between a ¾-in. steel header plate and ¾-in. O.D. copper tubes of 0.049-in. wall thickness. This boiler operates at 225 lb. per sq. in. and 395 deg. F. Other tests have shown no failures or evidence of flow in joints at 425 deg. F. under a load exceeding allowable creep limits of extra heavy copper tubing.

Effect of Design. By proper design it is always possible to make a joint as strong as the metals joined. Often the strength of the joint greatly exceeds that of the alloy employed for brazing. Strength of joints, therefore, depends upon the design, area and thickness of the joint, the composition of metals joined and of brazing alloy. It is not necessarily proportional to strength of brazing alloy. For these reasons data on strength of various brazing alloys and solders have been omitted from the accompanying table.

Conductivity. Joints having a conductivity equal to that of copper can be made if the recommended overlap of 1½ times section thickness is used. Such brazed joints retain their conductivity because they are not affected by oxidation or mechanical loosening.

Corrosion Resistance. Special silver brazing alloys are sometimes used to resist certain types of corrosion, but the common silver brazing alloys are adequately resistant to most forms of

corrosion for which copper-zinc and copper-nickel alloys are employed.

SILVER SOLDERS

While silver-lead solders are less important than brazing alloys and will remain so, current and probable future restrictions on tin will probably result also in wider use of silver-bearing solders, especially those based on lead. Such solders also contain cadmium and tin; they melt or flow between 560 and 715 deg. F.

MELTING RANGE. Silver solders may be characterized as "soft" but their melting range is well above that of ordinary "soft" tin-lead solders. Solders containing 2 to 6 percent silver have been applied primarily, heretofore, where they must withstand temperatures well above those at which common soft tin-lead solders melt, say above 400 deg. F.

COST. Even under "normal" conditions, a solder containing 2.5 percent silver and the remainder lead costs no more than one having approximately 25 percent tin and remainder lead.

STRENGTH. Although strength of joints made with solders containing 2.5 to 6 percent silver and the remainder largely lead or cadmium is much lower than is obtained with silver brazing alloys, such low-silver solders have the same strength as the common tin-lead soft solders at room temperatures and they retain considerable strength at temperatures that would melt tin-lead solders. The solder designated *O* in the table, for example, has been used successfully in electric motors in which heating caused tin-lead solders to fail.

Substitution of silver-lead solder for tin-lead in joining copper parts, avoids the reported tendency of tin-lead solder to form undesirable brittle copper-tin compound.

Silver Brazing Alloys and Silver-Bearing Solders

Designation	ASTM No.[a]	Composition, by Weight, Percent							Melting Point [d] (Solidus Temp.) deg. F.	Flow Point (Liquidus Temp.) deg. F.	Color	Use and Remarks
		Ag	Cu	Zn	Cd	Pb	Sn	Other				
A	1	10	52	38	(b)				1,510	1,600	Brass-yellow	Strongest joints heavy ferrous and non-ferrous; low price, high temp.
B	2	20	45	35	(b)				1,430	1,500	Brass-yellow	Joining copper-base alloys and steel parts.
C	3	20	45	30	5				1,430	1,500	Brass-yellow	General purpose for copper, brass, nickel, steel. Economical.
D	4	45	20	35					1,250	1,370	Yellow-white	Widest use in silver-copper-nickel group, ferrous, and non-ferrous. Free flowing.
E	5	50	34	16					1,280	1,425	Yellow-white	Much used on electrical and refrigerating equipment, ferrous and non-ferrous.
F	6	65	20	15					1,280	1,325	Silver-white	Silversmithing. Free flow color matches sterling.
G	7	70	20	10					1,325	1,390	Silver-white	Chiefly on sterling. Color matches sterling closely.
H	8	80	16	4					1,360	1,460	Silver-white	Copper wire for drawing. Is strong and ductile.
I	(p)	15	80					P-5	1,185	1,300	Yellow	Non-ferrous only. Free flow, ductile, moderate cost.
J	(p)	50	15.5	15.5	16			Ni-3	1,195	1,270	Yellow	Free flow, ferrous and non-ferrous. Carbide tips to steel.
K	(p)	50	15.5	16.5	18				1,160	1,175	Yellow	Lowest flow point of brazing alloys. Strong, ductile, ferrous and non-ferrous.
L		3.5 1.5				rem.[e]			579	653		Emergency alternate federal specification E-QQ-S-571A.
M		5–6				rem.			579	715		Electrical.
N		2.5				rem.			579	579		
O	(p)	2.5	0.25			97¼			580	661		Electrical, high temperature motors.
P		5			95				640	740		Strength at 400 deg. F., same as soft solders at room temperature.
Q		2				98			579	586		Electrical
R		5					95		330	563		
S	(p)	2				rem.	10	Bi 5, Sb 0.5	417	527	Silver-white	For many applications in which higher tin (soft) solders are now used.
T	(p)	1.5				rem.	15	Bi 5, Sb 0.5	417	496	Silver-white	For many applications in which higher tin (soft) solders are now used.
U	(p)	1.5				rem.	20	Bi 3, Sb 0.5	417	496	Silver-white	For many applications in which higher tin (soft) solders are now used.

Notes: (a) Max. allowable impurities in all ASTM silver solders is 0.15 percent. (b) Cadmium up to 0.50 percent max. is not considered a harmful impurity. (p) Proprietary alloy (not ASTM). (d) Technically, the melting point is actually the solidus temperature, but in shop practice, the flow point or liquidus temperature is often called the melting point. (e) rem. = remainder.

Alloys listed are only a few of scores classed as silver solders and silver brazing alloys, but are reasonably representative of types containing significant quantities of silver.

VINYLIDENE PLASTICS

Polyvinylidene chloride thermoplastics, trade-named "Saran" by Dow Chemical Company, have unusual chemical resistance and toughness. Petroleum and brine are basic raw materials. Extruded tubing is a good alternate for copper. Molded Saran parts can often be used in place of corrosion-resistant metals and hard rubber. Monofilaments with oriented molecular structures have high strength.

TUBING AND EXTRUDED SHAPES

EXTRUSION PROCESS. Saran can be extruded using modified screw type extrusion equipment. Designs permitting streamlined plastic flow are important. Metals which can be used in contact with Saran above 265 deg. F. in the heated sections of extrusion equipment are magnesium alloys, nickel, Z-nickel, Hastelloy B, and Stellite 10. Iron and copper-base alloys initiate decomposition of vinylidene chloride polymers at elevated temperatures.

The extruded plastic shape may be cooled and subsequently heat-treated. When first extruded the product is soft, weak and pliable. If allowed to remain at room temperature it will gradually harden while partially recrystallizing with a random crystal arrangement. By heat-treatment, recrystallization can be produced at faster controlled rates.

PHYSICAL PROPERTIES. The control of extrusion and heat-treatment of unoriented Saran permits a range of properties to be obtained. Tensile strengths can thus be controlled from 4,000 to 8,000 lb. per sq.in.; hardness from 65 to 95 Rockwell (Superficial 15Y); elastic elongations from 15 to 25 percent. Good fatigue life is illustrated by:

Specimens of $\frac{1}{16}$ in. tubing having a wall thickness of $\frac{1}{16}$ in., joined with "B" Parker standard couplings, when flexed through a 15-deg. angle 1,750 times per min. for 2,500,000 cycles, remained unruptured. In comparison, standard ¼-in. copper tubing failed at about 500 cycles in the same test.

Saran tubing retains its mechanical properties at elevated temperatures for short periods of time. For example, ¼ in. tubing with 0.035 in. wall showed no tendency to collapse within 3 hours at 300 deg. F. and 1 mm. absolute internal pressure. Tubing of the same size did not bulge or rupture when heated to 212 deg. F. and subjected to 200 lb. per sq.in. air pressure for 13 hr.

TUBING. Probably the most important application now is tubing. It is available in sizes ranging from ⅛ to ⅜ in. outside diameter with wall thickness ranging from 0.031 to 0.062 in. Its characteristics enable it to replace metal

TABLE I—Properties of Vinylidene Chloride

MECHANICAL PROPERTIES	UNORIENTED	ORIENTED
Tensile strength, lb. per sq. in.	4,000–8,000	15,000–40,000
Flexural strength, lb. per sq. in.	15,000–17,000	Flexible
Bursting strength of tubing	See Table II	
Elongation at yield point, percent	15–25	20–30
Modulus of elasticity, lb. per sq. in. x 10^5	0.7–2.0	0.7–2.0
Impact strength, ft. lb. per in., ½ x ½ in. notched bar, Izod	2–8	
Hardness, Rockwell Superficial 15Y	65–95	
Rockwell M	50–65	
Durometer A	Above 95	
Toughness and abrasion resistance	Excellent	Excellent

THERMAL PROPERTIES	UNORIENTED	ORIENTED
Thermal conductivity, cal. per sec. per sq. cm. per deg. C. per cm	0.00022	0.00022
Specific heat, cal. per deg. C. per g	0.32	0.32
Coefficient of thermal expansion, per deg. C.	15.8 x 10^5	
Resistance to heat, continuous, deg. F.	170	170
intermittant, deg. F.	212	Shrinks above 170
Heat distortion temperature, deg. F.	150–180	
Tendency to cold flow	Slight	
Softening point, deg. F.	240–280	240–280

ELECTRICAL PROPERTIES	UNORIENTED	ORIENTED
Volume resistivity, ohm per cm. (50 percent relative humidity and 25 deg. F.)	10^{14}–10^{16}	
Breakdown voltage, 60 cycles, instantaneous		
volts per mil at 1 mil thickness	3,000	
volts per mil at 20 mils thickness	1,500	
volts per mil at 125 mils thickness	500	
Dielectric constant		
at 60 cycles	3.0–5.0	3.0–5.0
at 10^3 cycles	3.0–5.0	3.0–5.0
at 10^6 cycles	2.5–5.0	2.5–5.0
Power factor		
at 60 cycles	0.03–0.08	0.03–0.08
at 10^3 cycles	0.03–0.15	0.03–0.15
at 10^6 cycles	0.03–0.05	0.03–0.05

CHEMICAL PROPERTIES	UNORIENTED	ORIENTED
Effect of weak acids	None	None
Effect of strong acids	Darkens in H_2SO_4; others, none	
Effect of weak alkalies	None	None
Effect of strong alkalies	Affected by NH_4OH; darkens in caustic; others, none	
Effect of organic solvents	Highly resistant	Highly resistant
Effect of metal inserts	None	
Effect of age	None	None
Effect of sunlight	Darkens slightly	Darkens slightly

OPTICAL PROPERTIES	UNORIENTED	ORIENTED
Refractive index, *nD*	1.60–1.63	1.60–1.63
Clarity	Translucent to opaque	
Color possibilities	Extensive	Extensive

GENERAL PROPERTIES	UNORIENTED	ORIENTED
Specific gravity	1.68–1.75	1.68–1.75
Specific volume, cu. in. per lb	15.8–16.16	15.8–16.16
Water absorption (A.S.T.M. Test No. D-570–40-C)		
percent after 24 hr. at 25 deg. C.	0.00	0.00
percent after 168 hr. at 25 deg. C.	0.05	0.05
Water permeability	Negligible	Negligible
Machining qualities	Good	
Burning rate	Self extinguishing	Self extingushing
Odor, taste, toxic effect	None	None

TABLE II—Bursting Strength of Vinylidene Chloride Tubing

Outside Diameter, in.	Wall Thickness, in.	Bursting Pressure, lb. per sq. in.	Working Pressure (Safety factor = 5) lb. per sq. in.
1/4	0.060	1,840	370
1/4	0.045	1,260	250
1/4	0.030	630	130
3/16	0.033	1,150	230
5/32	0.031	1,290	260
1/8	0.030	1,795	340

Note: These strengths are at room temperature. Charts are avilable for bursting strengths at higher or lower temperatures.

TABLE III — Molding Qualities of Vinylidene Chloride

COMPRESSION MOLDING	
Temperature, deg. F	220–350
Pressure, lb. per sq. in	500–5,000
INJECTION MOLDING	
Temperature, deg. F	300–400
Pressure, lb. per sq. in	10,000–30,000
Mold Shrinkage, in. per in	0.008–0.012
EXTRUSION temperature, deg. F	Up to 375

tubing in many applications except where high temperatures and high pressures are encountered. Bursting strengths at room temperature are given in Table II, and drop off considerably at higher temperatures. The tubing can be joined by special Saran injection molded fittings, Parker standard tube couplings, S.A.E. and other standard flare type of compression fittings. At room temperature standard flaring equipment can be used. At temperatures as low as 50 deg. F., dip ends of tubes in hot water and use heated flaring tools. Tube can be trimmed with knife, scissors or file.

TYPICAL APPLICATIONS of tubing include oil and gasoline lines for engines and machines; tubing for recording devices and gages; siphon tubes and lead-in pipes for chemicals; refrigerant transfer tubes; electrical insulation, air and water lines. Applications for other extruded shapes include: rods for making gaskets, valve seats; medicinal probes; chemically resistant flexible tubing and pipe, tape for wrapping joints; chemical conveyor belts; tape and strips for die cutting; and various items of wearing apparel. It is available in sheets ranging from $\frac{1}{64}$ to $\frac{1}{2}$ in. thick.

INJECTION MOLDED VINYLIDENE CHLORIDE

MOLDING TECHNIQUE. Here as in extruding, equipment consists of modified standard injection molding machine. Conventional injection molding die design, die metals, and parts design considerations apply. Molding properties are shown in Table III.

Injection molding dies are not normally operated cold unless soft, flexible, amorphous pieces are desired. Rapid hardening of Saran is accomplished by heat-treatment producing recrystallization. Parts are ejected from heated dies at temperatures as high as 70 to 90 deg. C., in a strain-free, warp-free, dimensionally stable form. This promotes rapid cycles with heavy sections.

PHYSICAL PROPERTIES. Physical properties of molded Saran are subject to the same considerations as described for extruded Saran.

TYPICAL APPLICATIONS. Many applications of molded Saran have been to replace such strategic materials as nickel, aluminum, stainless steel, and rubber in applications requiring chemical resistance. Examples are:

Spray gun handles. Saran here replaced aluminum because of solvent and abrasion resistance.

Valve seats. Good abrasion resistance of Saran reduces gullying. Resilience improves seating qualities. Resistance to corrosive gases and liquors is sometimes an advantage.

Moldings for rayon industry. Spinnerette couplings, gasket holders, filter parts, nozzle tips, rollers, guides, are now made of Saran instead of hard rubber. Factors in choice are its inertness and stability in contact with chemicals and solvents.

HIGH-STRENGTH FILAMENTS AND TAPES

ORIENTATION TECHNIQUE. A high-strength form of Saran is produced by extrusion, subsequent stretching and heat-treatment. During mechanical stretching there is a partial recrystallization and orientation of the crystallites along the major axis of the strand. The resulting tensile strength is a function of elongation as illustrated by Fig. 1.

PHYSICAL PROPERTIES. The uni-directional properties of high tensile strength, great flexibility, long fatigue life, and good elasticity are particularly desirable for small monofilament sections where the load is along the longitudinal axis. Tensile strengths of Saran monofilament vary with the temperature, as shown in the chart Fig. 2.

TYPICAL APPLICATIONS. Extruded and oriented sections ranging in size from 0.007 to 0.060 in. dia. monofilaments, are now being produced for textiles, and in other shapes having maximum dimensions up to 0.200 in. These materials can be fabricated by braiding, weaving, knitting and twisting. Many present applications are in fields formerly supplied by imported natural products, such as hemp, long fiber paper, reed, rattan, horsehair, Spanish silkworm gut, and linen. Typical examples include special ropes and fabrics which call for high wet strength, flexibility, abrasion and chemical resistance, and ease of cleaning.

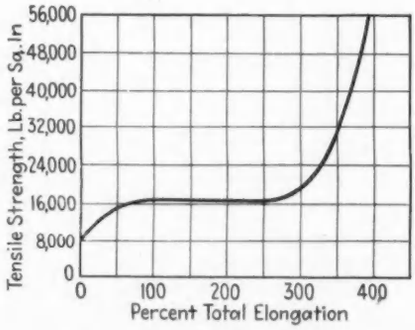


Fig. 1—Effect of percent total elongation of vinylidene chloride monofilaments upon tensile strength. Stretching time is 5 min. for this curve

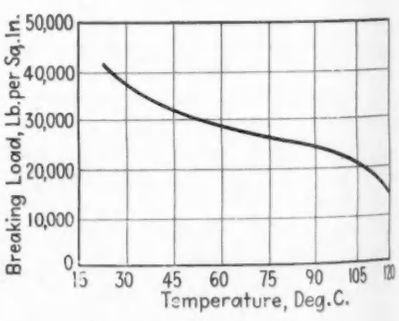


Fig. 2—Tensile strength of vinylidene chloride oriented monofilaments is reduced at higher temperatures, as shown by this chart of 0.027 in. dia. cordage

NEOPRENE RUBBER

One of the so-called synthetic rubbers, neoprene, has the advantage of an accumulation of experience in many widely diversified applications because of its early introduction to industry. Like rubber and other synthetic rubbers, it is produced with a wide range of physical properties, depending on the kind and amount of compounding ingredients, and on its processing. Chemically, neoprene is polymerized chloroprene having the same structure as isoprene, primary constituent of natural rubber, but differing by substitution of a chlorine atom for a methyl group, CH_3. In butabiene, basis for Buna rubbers, a hydrogen atom replaces the methyl group in the molecule.

SPECIFIC GRAVITY. Specific gravity of the pure gum is 1.25, but its compounds because of incorporation of pigments range from 1.40 to 1.80. Heavily pigmented mixes may have a specific gravity as high as 3.0.

TENSILE STRENGTH. Comparable with like compositions of natural rubber. Compounds may be formulated having an ultimate tensile strength of 200 to 4,500 lb. per sq.in.

TEAR STRENGTH. Varies with type of compound, but, in general, is slightly lower than for rubber.

ABRASION RESISTANCE. Similar to rubber in ordinary nondeteriorating service conditions. Best evaluation determined by test installations. Accelerated wear test in the presence of oil, using high-speed band saw, wore rubber tire tread more than twice as fast as neoprene during 5-min. contact.

ELASTICITY. Elasticity and rebound rated good, only slightly inferior to natural rubber. Rate and degree of return to original dimensions are functions not only of degree of distortion but of time and temperature. Neoprene can be compounded with elasticity superior to rubber under certain conditions.

HYSTERESIS. Accompanying compression loading and unloading curves, Fig. 1, show difference between neoprene and rubber and indicate rebound characteristics. Area between loading and unloading curves is measure of ability to absorb vibration. Difference is largely in unloading curves. Compounds with same hysteresis as natural rubber can be made if desired.

VIBRATION DAMPING. Since neoprene does not recover as rapidly as rubber it has an advantage in absorbing vibrations but may require change in mounting design to obtain benefits. Under dynamic loads, neoprene compounds show more rapid damping effect than rubber compositions which have the same load-deflection rates. In tests the number of oscillations produced by identical shock loads was 10 for neoprene as compared with 16 for rubber.

PERMANENT SET. Neoprene exhibits, like rubber, a drift or gradual deformation under constant load. However, deflection soon reaches maximum and permanent distortion upon release is low. Recovery is almost as complete as for rubber compositions, but is slower.

Significance of the comparison in Table II is that although recovery of neoprene stock is slower, it is of same order of magnitude. If drift is important factor in design, sufficient allowance should be made for it. It is slightly greater than that for rubber.

HEAT RESISTANCE. All neoprene compounds are superior to rubber in heat resistance but best results are obtained with special type of neoprene. These show greatest superiority between 180 and 300 deg. F. Heat resisting neoprenes gradually harden at 300 deg. F. and up, and finally fail by becoming brittle. Neoprene is not a thermoplastic and hence does not lose resilience at elevated temperatures. Unless highly compounded with oils or organic diluents they do not support combustion as do rubbers. There are many applications where fire prevention is important.

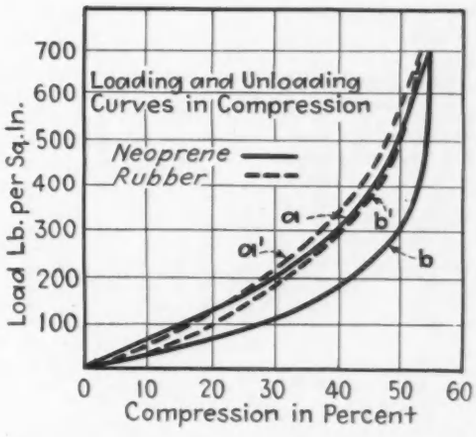


Fig. 1—Loading and unloading curves under compression

Table I—Damping Characteristics of Neoprene and Rubber

Compression Modulus* Lb. per sq. in	Durometer Hardness	Percent Resilience	
		Rubber	Neoprene
150	25	91	78
340	35	92	73
530	45	89	81
720	55	82	78
1,100	65	74	70

*Rubber and neoprene compounds of similar hardness have about the same compressive modulus.

Table II—Percent Compression Set

Disks ¾ in. dia., ½ in. thick

Time after load release	Neoprene	Rubber
Immediately	16.6	8.3
After 30 min	12.5	7.9
After 24 hr	10.5	7.5

(Continued on next page)

COLD RESISTANCE. A new type of neoprene that hardens less at sub-zero temperatures than does natural rubber is now available. This was not true of older neoprenes, but these would regain original physical properties when mechanically worked even when low temperatures were maintained. These compounds had greater freeze resistance after absorbing a light petroleum lubricant.

AGING. Neoprene is many times more resistant to oxidation than the best modern rubber compounds. Presence of ozone in air has no effect upon neoprene. Accompanying oxygen bomb test data, Fig. 2, cannot be interpreted into years of expected life but do give reliable comparison in service. Similar results have been obtained with other types of compounds.

Aging caused by constant flexing also shows the excellent resistance of neoprene to flex cracking.

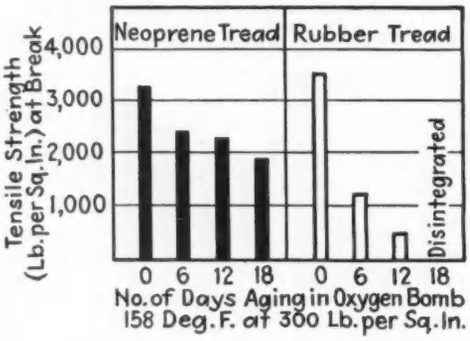


Fig. 2—Oxygen bomb tests cannot be interpreted into expected life but do give a fair service comparison

SUNLIGHT RESISTANCE. Practically unaffected by sunlight. Only change is slight stiffening after long exposure, which contracts with severe cracking of rubber, particularly when test specimens are flexed or put under strain during tests. Effect of sunlight upon diffusion rate of hydrogen after 7 months exposure was an increase of 9 percent as compared with 1,471 percent for a rubber film.

THERMAL CONDUCTIVITY. Although classed as a heat insulator neoprene has 28 percent greater conductivity than rubber. Based on "pure gum" compound, coefficient of thermal conductivity is 1.45 B.t.u. per hr. per sq.ft. per in. thickness for a temperature of 1 deg. F.

ELECTRICAL PROPERTIES. Neoprene compounds are fair electrical insulators but inferior to similar rubber compounds in power factor and volume resistivity, which may have a resistivity of 7.5×10^{12} ohm-cm. at 77 deg. F. Break-down strength of neoprene compounds is slightly superior to that of rubber compounds. A "pure gum" or high neoprene-containing compound will withstand 860 volts per mil of thickness when dry. Neoprene compounds have excellent resistance to corona.

When dry at 77 deg. F. neoprene compounds designed for best insulating value have approximately the following electrical characteristics:

D.C. volume resistivity	7×10^{12} ohm-cm.
Dielectric constant at 1,000 cycle per sec.	7.5 to 14.4
Power factor at 1,000 cycles per sec.	1.07 to 6 percent
Dielectric strength RMS at 60 cycles per sec.	800 volts per mil

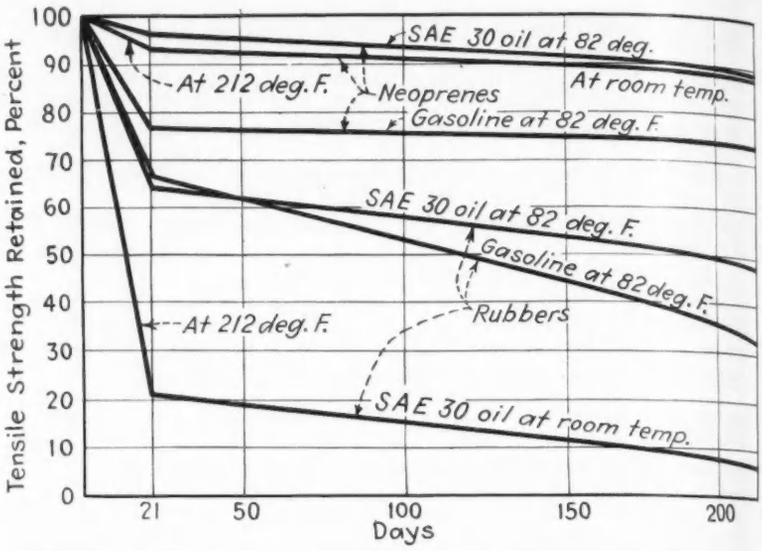


Fig. 3—Effect of immersion in gasoline and lubricating oil on tensile strength of neoprene and rubber

OIL RESISTANCE. Neoprene swells from contact with oils, but to a much less extent than rubber. More important it retains a considerable portion of its strength, toughness and elasticity while swollen. Chloroprene rubber can be compounded so as to undergo no change in dimensions in any given oil at any specified temperature, but such a non-swelling compound is frequently not the most serviceable one that can be produced. A new low swell neoprene is also available.

Amount of volume increase varies considerably with different crude petroleum oils and refined products. Temperature is also an important factor. Neoprene vulcanizates reach equilibria with respect to swelling or volume increase when immersed in lubricating oils.

Asphaltic base mid-continent and coastal crudes and refined products cause more deterioration than paraffin base crudes from Pennsylvania. Heavier fractions of refined crude oil have less effect than lighter ones.

Because progress has been rapid in recent months commercial products will not swell anywhere near as much today as past experience indicates.

VEGETABLE OILS. Neoprene is more resistant than rubber to all vegetable oils. Although most vegetable oils have little effect on neoprene, a few cause marked swelling, but without serious deterioration. Since a few of these oils will render neoprene unsuitable in certain services, tests are advised before extensive installations.

ANIMAL FATS AND FISH OILS. Neoprene is much more resistant than rubber to lard, butterfat and fish oils. Human perspiration also has greatly reduced effects on neoprene as compared with rubber.

RESISTANCE TO CHEMICALS. Most salts have practically no effect. Sulphuric and hydrochloric acids up to 30 percent concentration cause no serious deterioration even at high temperatures. Higher concentrotions of sulphuric acid have been handled with special compounds. Hydrofloric acid causes comparatively slow deterioration. Alkalies cause little trouble in ordinary concentrations. Neoprene is not recommended for use with concentrated nitric acid or with chromic acid.

"Freon" refrigerants, methyl chloride and sulphur dioxide in both liquid and gas phases have much less effect on neoprene than on rubber. Both are stable in presence of ammonia.

Halogens and halogen-substituted hydrocarbons, such as chlorine, bri-

mine and iodine have little effect on neoprenes, in contrast with rubber. High concentrations cause formation of thin, hard, protective film on neoprene surface, which protects inside mass. Carbon tetrachloride causes swelling and deterioration of both rubber and neoprene. Neoprene is superior.

Of the organic chemicals, alcohols have little effect on either neoprene or rubber, although contamination of the alcohols by leaching may preclude use of either. Aromatic hydrocarbons such as benzene, toluene and xylene attack both. While neoprenes retain rubber-like properties to a greater extent, they are not good enough for the great majority of such uses. Esters, such as amyl acetate, actively deteriorate neoprene.

For particularly low water absorption, special compounds are required although, like rubber, neoprene is a water-proofing material.

Table III—Effect of Immersion in Kerosene on Neoprene and Rubber Compounds

COMPOUND	NEOPRENE				RUBBER		
	A	B	C	D	E	F	G
ORIGINAL PROPERTIES							
Tensile strength, lb. sq. in.	2,275	1,425	2,375	1,350	2,400	2,400	2,350
Elongation at break, percent	370	700	280	650	240	260	290
Tear resistance	372	395	329	192	210	292	385
Hardness	73	73	72	32	80	79	72
AFTER 21 DAYS' IMMERSION IN KEROSENE AT 82 DEG. F.							
Tensile strength	1,680	1,115	1,795	1,225	710	525	630
Percent deterioration in tensile	26.2	21.5	24.4	9.3	70.4	78.1	73.2
Elongation at break	330	700	230	600	160	140	160
Tear resistance	137	186	105	185	43	44	40
Hardness	50	52	52	39	51	49	46
Percent volume increase	31.4	31.8	30.6	2.1	121	126	140
AFTER 21 DAYS' IMMERSION IN KEROSENE AT 212 DEG. F.							
Tensile strength	850	705	890	870	Too weak to test		230
Percent deterioration in tensile	62.6	50.5	62.5	35.6	" " "		90.1
Elongation at break	240	180	200	310	" " "		....
Tear resistance	74	51	47	99	" " "		19
Hardness	33	40	33	38	" " "		....
Percent volume increase	49.9	51.7	49.3	7.6	368	492	383

Table IV—Swell of Rubber and Neoprene Compounds in Various Oils Immersed 21 days at 82 deg. F.

Type of Oil	Rubber percent	Neoprene percent	Type of Oil	Rubber percent	Neoprene percent
Butter	68.1	10.9	Tung oil	24.0	13.4
*Lard	59.6	4.7	*Soya bean oil	59.6	9.1
*Cod Liver oil	61.7	11.4	Olive oil	64.1	7.6
*Cocoanut oil	66.5	11.2	Rosin oil	96.1	12.2
Oleic acid	95.0	48.2	Turpentine	194.1	82.3
Linseed oil	62.3	16.4			

*Further tests at 212 deg. F. showed superiority of neoprene with respect to retention of tensile strength, hardness, and volume increase.

DIFFUSION OF GASES. Many gases that readily permeate a rubber film can be retained by neoprene. Fabric covering power of neoprene is less than rubber and it is more difficult to obtain a continuous film on cloth. On a weight basis and on freshly vulcanized samples permeability of neoprene is about one-third to one-fourth that of rubber. After seven months, permeability of hydrogen through rubber-impregnated fabric was 150 times that of neoprene fabric.

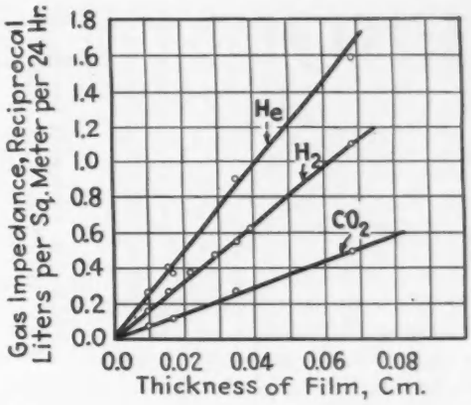


Fig. 4—Relation of thickness of neoprene films to gas impedance

FABRICATED PARTS. Production of vulcanized products requires extensive rubber making equipment. However, neoprene cements, obtained by dissolving unvulcanized neoprene in a solvent, can be employed by almost any manufacturer. Another neoprene product that can be used by manufacturers in general is neoprene latex, or colloidal neoprene in water. This is used for impregnating porous materials or may be deposited as a film on smooth surfaces.

Molded parts may be made in even more intricate shapes than plastics but the molding methods vary widely. Neoprene parts are usually supplied in the form and shape required for use and do not require further cutting or fabrication other than assembling.

ADHESION TO METALS. Excellent. Compares with rubber in this respect.

ODOR. An odorless type neoprene has been developed. It has slightly superior physical properties to standard neoprene and retains same resistance to deterioration from exposure to heat, oils and chemicals.

SULPHUR CONTENT. Neoprene does not require sulphur for vulcanization although it may be incorporated to impart certain characteristics. Sulphurless compounds are of value for use in contact with materials that are attacked by sulphur or where adhesion to metal surfaces must be avoided.

COMPOUNDING LIMITATIONS. Neoprene is not mixed with rubber. It can be but rubber reduces resistance to deterioration and other fillers are cheaper. No known mixture of neoprene with compound ingredients can be made to form a vulcanizate having properties corresponding to those of ebonite or "hard rubber". Same degree of hardness may be approximated by stiffening the compounds with large amounts of fillers or reinforcing pigments, but general properties of these combinations do not correspond to those of ebonite.

PROPERTIES of MUNTZ METAL

R. A. WILKINS and E. S. BUNN
Revere Copper and Brass Incorporated

MUNTZ METAL, an alloy of copper and zinc containing 60 percent copper, is used extensively where reasonably good resistance to corrosion in combination with high strength and moderate ductility are required. Applications of this class are boat and piling sheathing, tube sheets and baffle and support plates in heat exchangers. The name is traced to a patent issued in 1832 to Sir Frederick Muntz, Muntz Metal Company in England.

The accompanying charts give mechanical properties of cold worked and annealed strip, rod and tubing for ready-to-finish grain sizes 0.015 and 0.045 mm., which represent limits commonly encountered in commercial annealing.

The cold working series for rod is based on material that has been extruded at 1,250 deg. F., producing a grain size of 0.010 mm.

As would be expected, these charts show that mechanical properties are markedly influenced by ready-to-finish grain size, especially the cold worked series shown in Figs. 1 to 4. This range is typical of commercial production and as such is recognized in commercial specifications, control of ready-to-finish grain size from one lot to another for a given amount of cold work is essential in securing identical properties in Muntz metal strip.

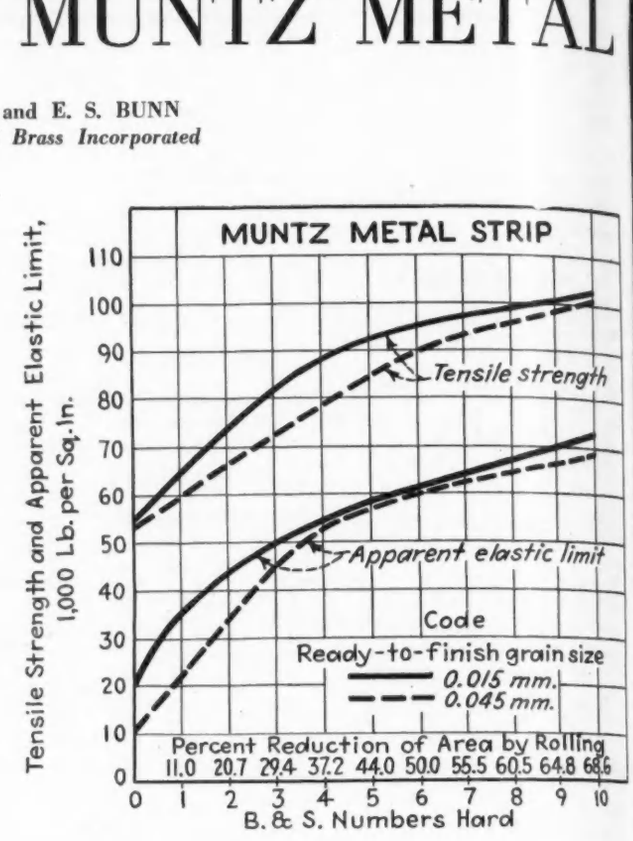

MUNTZ METAL STRIP
Tensile Strength and Apparent Elastic Limit, 1,000 Lb. per Sq. In.
Tensile strength
Apparent elastic limit
Code
Ready-to-finish grain size
0.015 mm.
0.045 mm.
Percent Reduction of Area by Rolling
11.0 20.7 29.4 37.2 44.0 50.0 55.5 60.5 64.8 68.6
B. & S. Numbers Hard

Fig. 1—Effect of cold work on tensile strength and apparent elastic limit of strip previously annealed to grain sizes of 0.015 mm. and 0.045 mm.

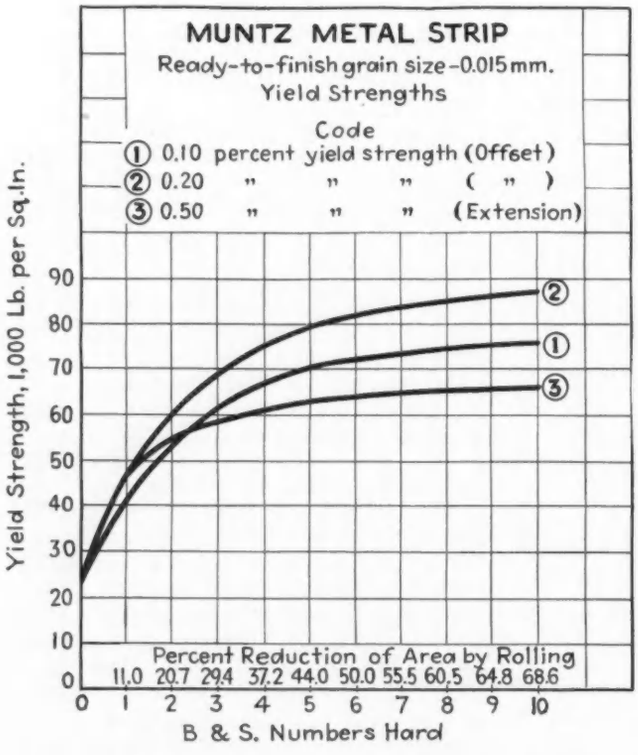

MUNTZ METAL STRIP
Ready-to-finish grain size-0.015 mm.
Yield Strengths
Code
① 0.10 percent yield strength (Offset)
② 0.20 " " " (")
③ 0.50 " " " (Extension)
Yield Strength, 1,000 Lb. per Sq. In.
Percent Reduction of Area by Rolling
11.0 20.7 29.4 37.2 44.0 50.0 55.5 60.5 64.8 68.6
B & S. Numbers Hard

Fig. 2—Effect of cold work on yield strengths of strip previously annealed to a grain size of 0.015 mm.

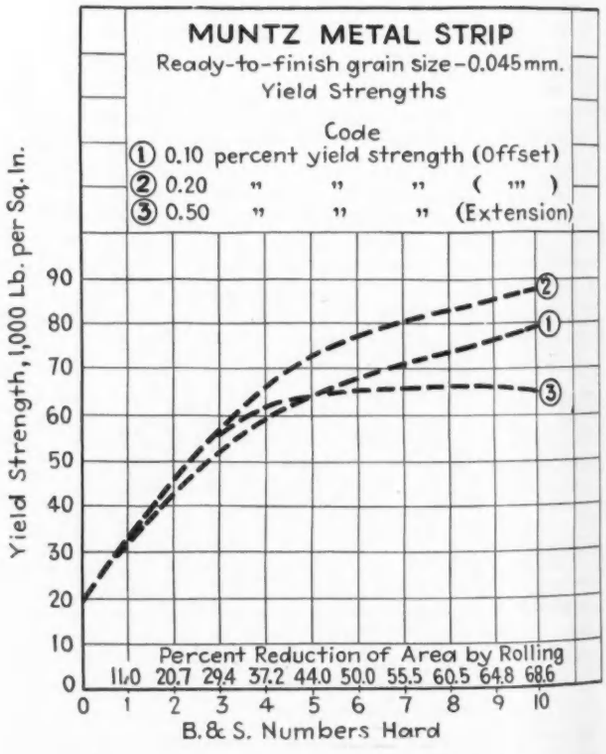

MUNTZ METAL STRIP
Ready-to-finish grain size-0.045 mm.
Yield Strengths
Code
① 0.10 percent yield strength (Offset)
② 0.20 " " " (")
③ 0.50 " " " (Extension)
Yield Strength, 1,000 Lb. per Sq. In.
Percent Reduction of Area by Rolling
11.0 20.7 29.4 37.2 44.0 50.0 55.5 60.5 64.8 68.6
B. & S. Numbers Hard

Fig. 3—Effect of cold work on yield strengths of strip previously annealed to a grain size of 0.045 mm.

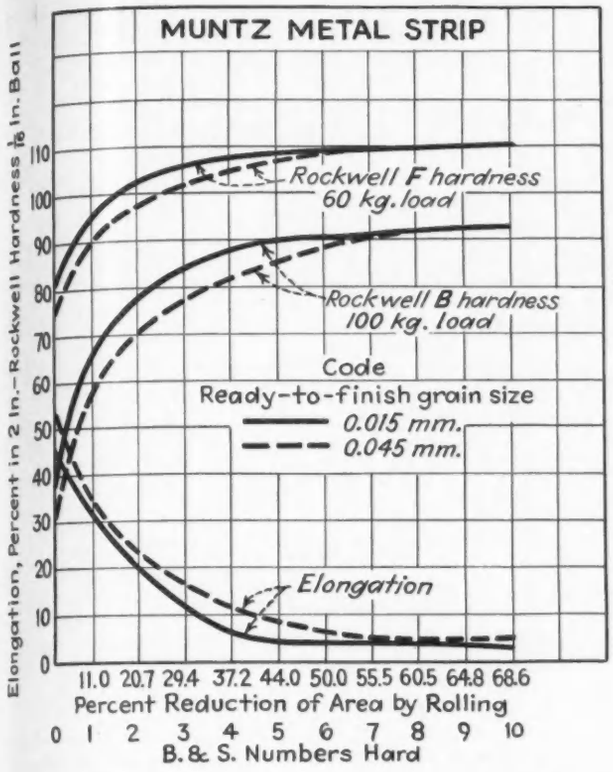

MUNTZ METAL STRIP
Elongation, Percent in 2 In.–Rockwell Hardness 1/16 In. Ball
Rockwell F hardness
60 kg. load
Rockwell B hardness
100 kg. load
Code
Ready-to-finish grain size
0.015 mm.
0.045 mm.
Elongation
11.0 20.7 29.4 37.2 44.0 50.0 55.5 60.5 64.8 68.6
Percent Reduction of Area by Rolling
0 1 2 3 4 5 6 7 8 9 10
B. & S. Numbers Hard

Fig. 4—Effect of cold work on Rockwell hardness and elongation of Muntz metal (60.50 percent copper) strip previously annealed to grain sizes of 0.015 mm. and 0.045 mm.

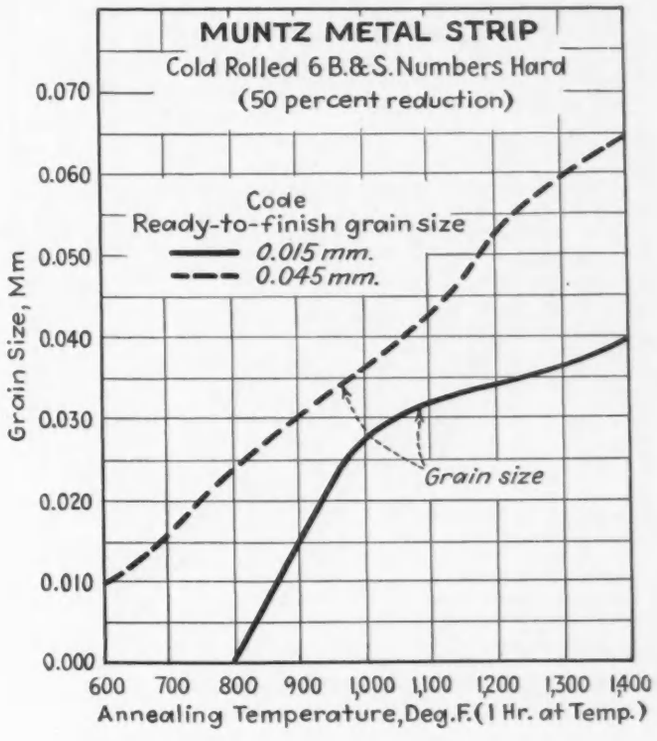


Fig. 5—Effect of annealing on grain size of strip previously cold worked 6 B & S numbers (50 percent reduction of area) from grain sizes of 0.015 mm. and 0.045 mm.

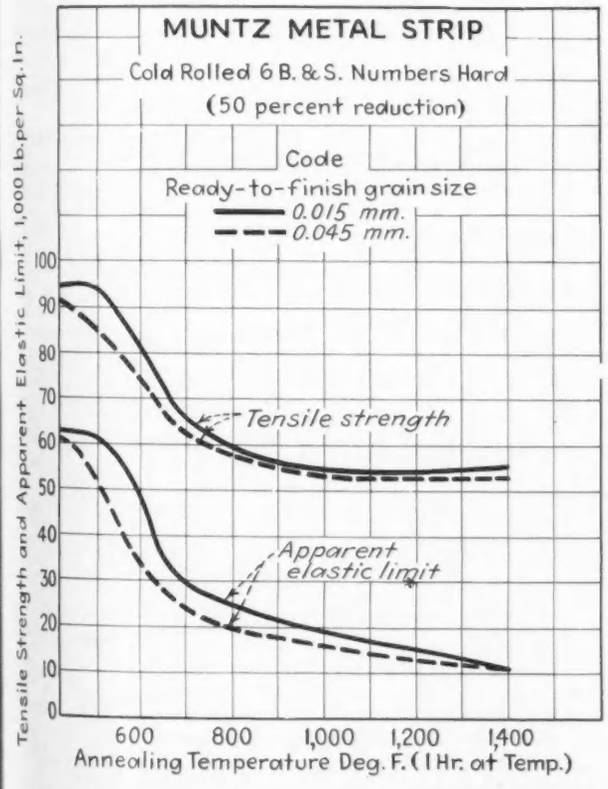

MUNTZ METAL STRIP
Cold Rolled 6 B. & S. Numbers Hard
(50 percent reduction)
Code
Ready-to-finish grain size
0.015 mm.
0.045 mm.
Tensile Strength and Apparent Elastic Limit, 1,000 Lb. per Sq. In.
Tensile strength
Apparent elastic limit
Annealing Temperature Deg. F. (1 Hr. at Temp.)

Fig. 6—Effect of annealing on tensile strength and apparent elastic limit of strip previously cold worked 6 B & S numbers (50 percent reduction of area) from grain sizes of 0.015 mm. and 0.045 mm.

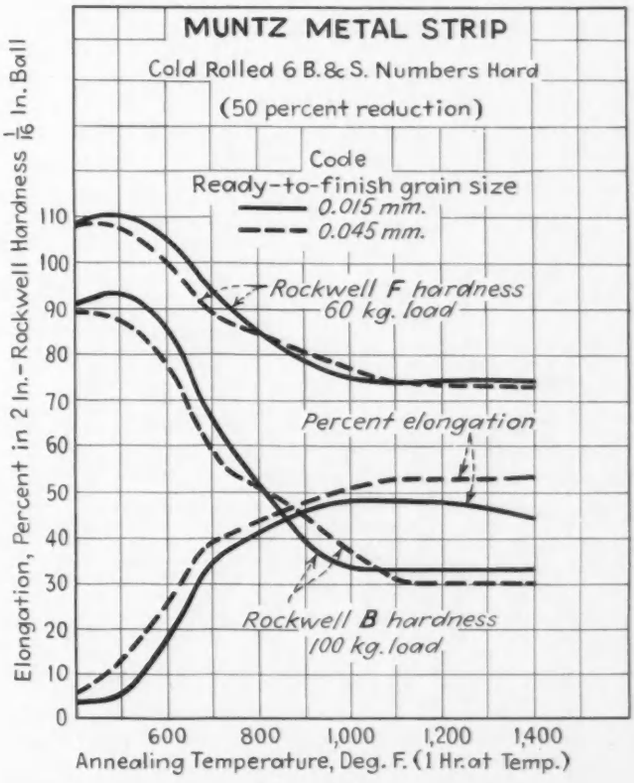


Fig. 7—Effect of annealing on Rockwell hardness and elongation of strip previously cold worked 6 B & S numbers (50 percent reduction of area) from grain sizes of 0.015 and 0.045 mm.

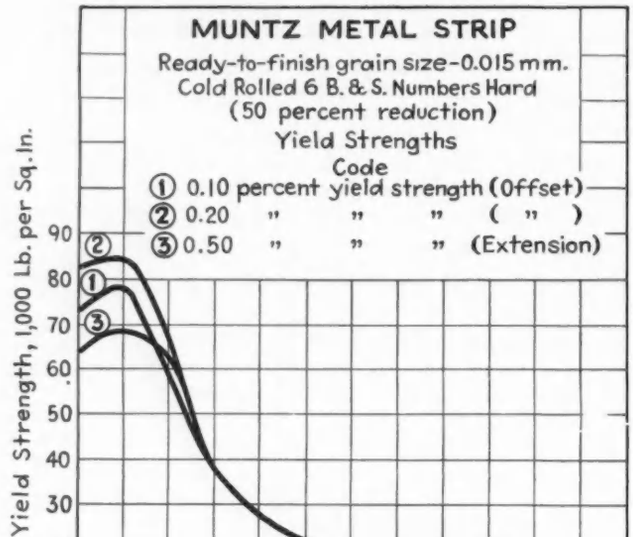

MUNTZ METAL STRIP
Ready-to-finish grain size-0.015 mm.
Cold Rolled 6 B. & S. Numbers Hard
(50 percent reduction)
Yield Strengths
Code
① 0.10 percent yield strength (Offset)
② 0.20 " " " (")
③ 0.50 " " " (Extension)
Yield Strength, 1,000 Lb. per Sq. In.
Annealing Temperature Deg. F. (1 Hr. at Temp.)

Fig. 8—Effect of annealing on yield strength of strip previously cold worked 6 B & S numbers (50 percent reduction of area) from a grain size of 0.015 mm.

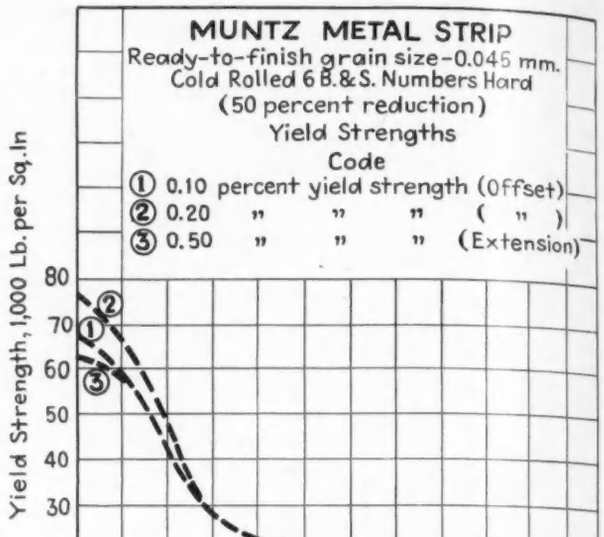

MUNTZ METAL STRIP
Ready-to-finish grain size-0.045 mm.
Cold Rolled 6 B.&S. Numbers Hard
(50 percent reduction)
Yield Strengths
Code
① 0.10 percent yield strength (Offset)
② 0.20 " " " (")
③ 0.50 " " " (Extension)
Yield Strength, 1,000 Lb. per Sq. In
Annealing Temperature, Deg. F. (1 Hr. at Temp.)

Fig. 9—Effect of annealing on yield strength of strip previously cold worked 6 B & S numbers (50 percent reduction of area) from a grain size of 0.045 mm.

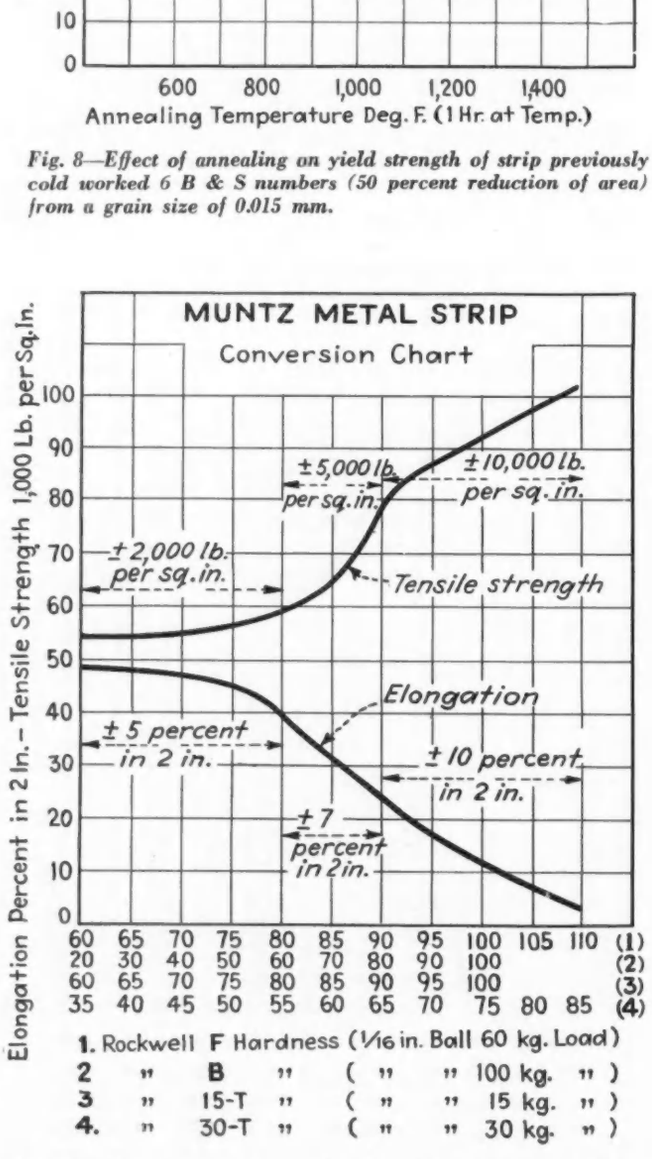

MUNTZ METAL STRIP
Conversion Chart
Elongation Percent in 2 In. – Tensile Strength 1,000 Lb. per Sq. In.
± 2,000 lb. per sq. in.
± 5,000 lb. per sq. in.
± 10,000 lb. per sq. in.
Tensile strength
Elongation
± 5 percent in 2 in.
± 10 percent in 2 in.
± 7 percent in 2 in.
1. Rockwell F Hardness (1/16 in. Ball 60 kg. Load)
2 " B " (" " 100 kg. ")
3 " 15-T " (" " 15 kg. ")
4. " 30-T " (" " 30 kg. ")

Fig. 10—This chart can be employed to determine approximate tensile strength and elongation of strip when only Rockwell hardness is known. It is accurate for thicknesses between 0.020 and 0.080 in. within the given limits

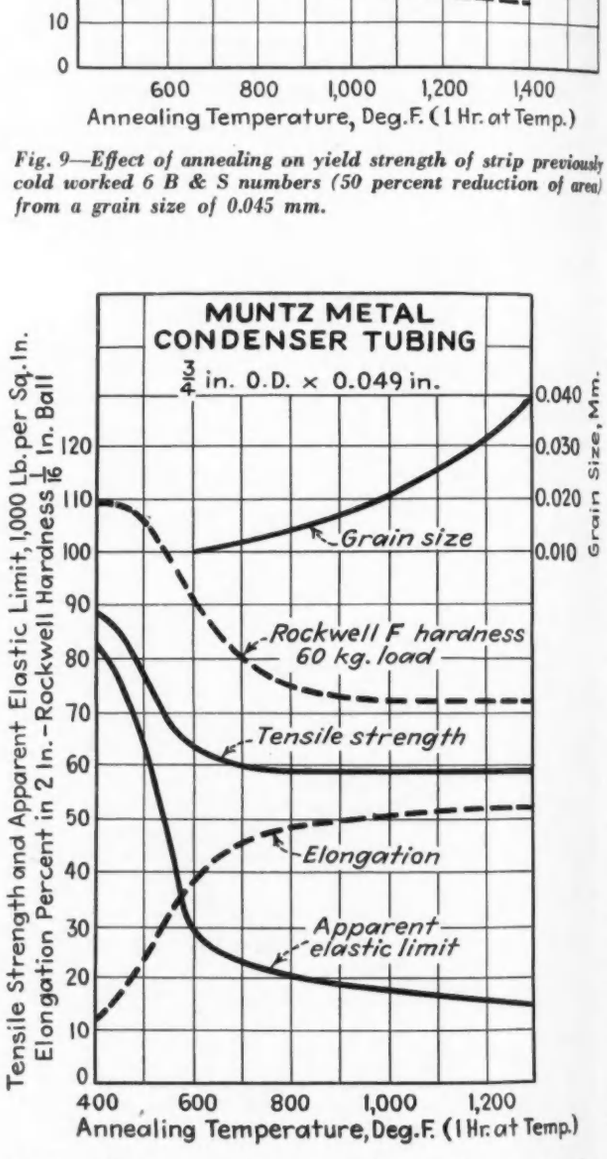

MUNTZ METAL CONDENSER TUBING
3/4 in. O.D. x 0.049 in.
Tensile Strength and Apparent Elastic Limit, 1,000 Lb. per Sq. In.
Elongation Percent in 2 In. – Rockwell Hardness 1/16 In. Ball
Grain Size, Mm.
Grain size
Rockwell F hardness 60 kg. load
Tensile strength
Elongation
Apparent elastic limit
Annealing Temperature, Deg. F. (1 Hr. at Temp.)

Fig. 11—Effect of annealing on tensile strength, apparent elastic limit, Rockwell hardness, elongation and grain size of Muntz metal (60.80 percent copper) tubing previously cold worked 50 percent from a grain size of 0.020 mm.

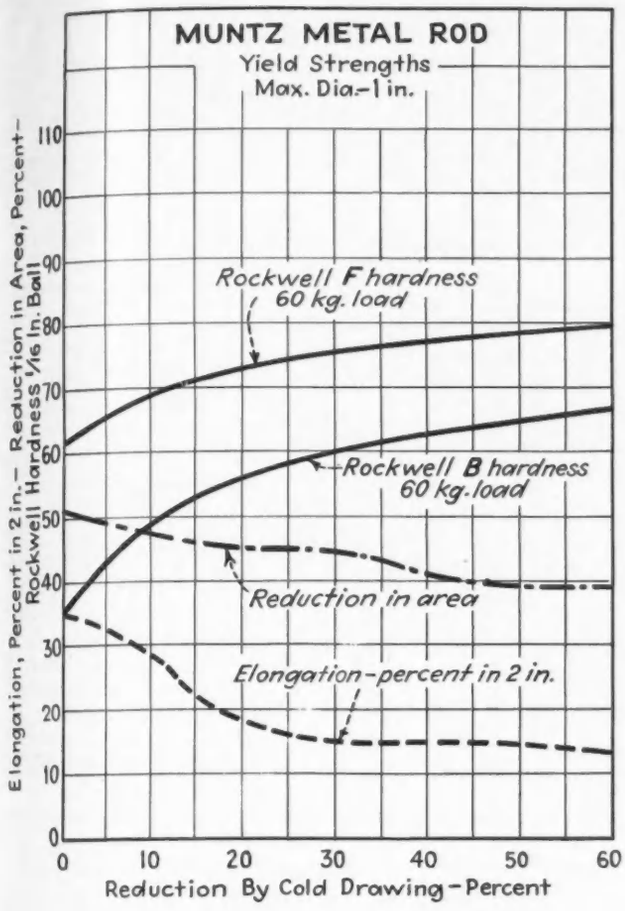


Fig. 12—Effect of cold work on Rockwell hardness, percent elongation and reduction of area of Muntz metal rod previously extruded to a grain size of 0.010 mm.

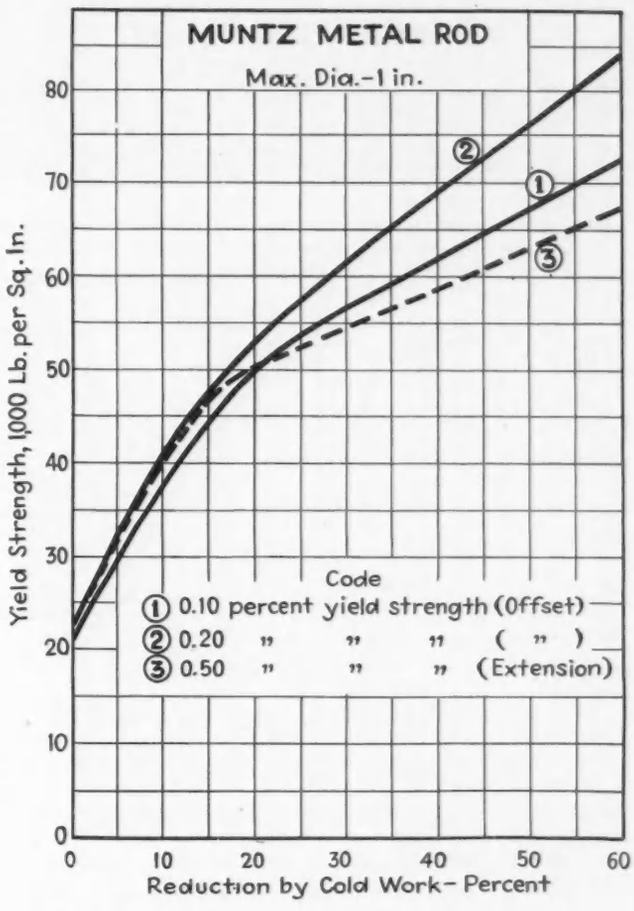

MUNTZ METAL ROD
Max. Dia.-1 in.
Code
① 0.10 percent yield strength (Offset)
② 0.20 " " " (")
③ 0.50 " " " (Extension)
Yield Strength, 1,000 Lb. per Sq. In.
Reduction by Cold Work- Percent

Fig. 13—Effect of cold work on yield strength of Muntz metal rod previously extruded to a grain size of 0.010 mm.

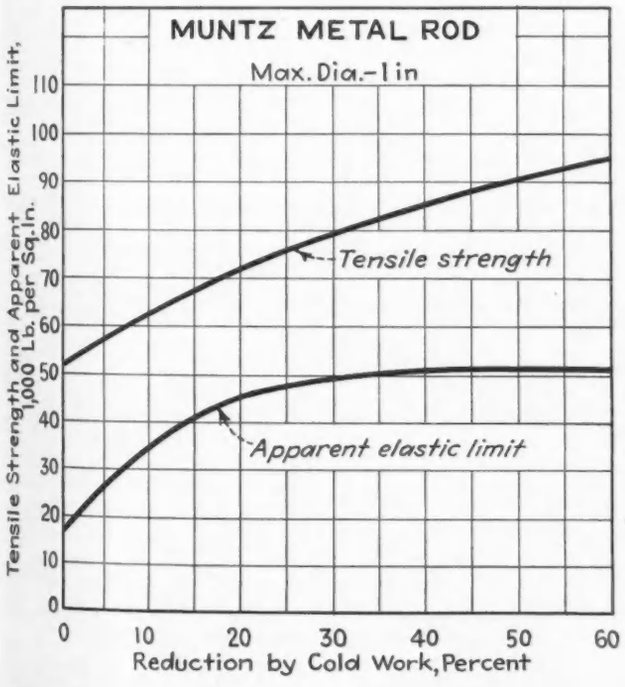

MUNTZ METAL ROD
Max. Dia.-1 in
Tensile strength
Apparent elastic limit
Tensile Strength and Apparent Elastic Limit, 1,000 Lb. per Sq. In.
Reduction by Cold Work, Percent

Fig. 14—Effect of cold work on tensile strength and apparent elastic limit of Muntz metal (60.73 percent copper) rod previously extruded to a grain size of 0.010 mm.

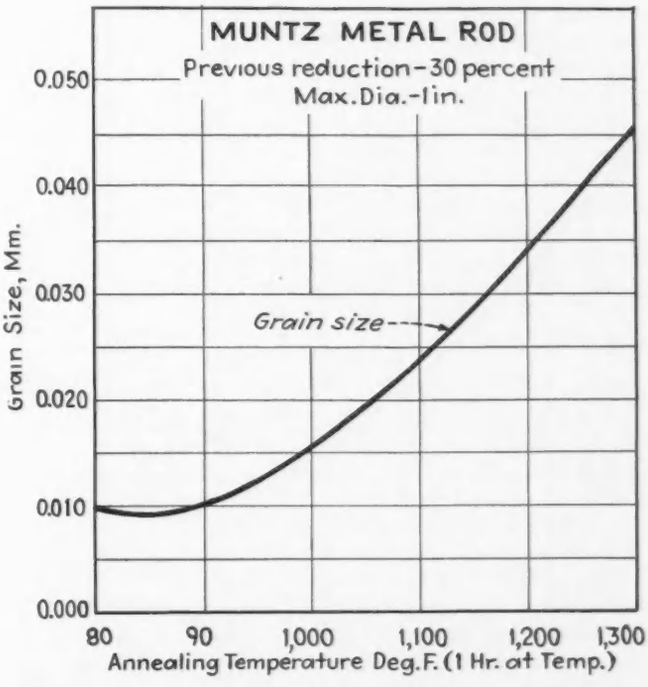


Fig. 15—Effect of annealing on grain size of rod previously cold worked to 30 percent reduction of area from a grain size of 0.010 mm.

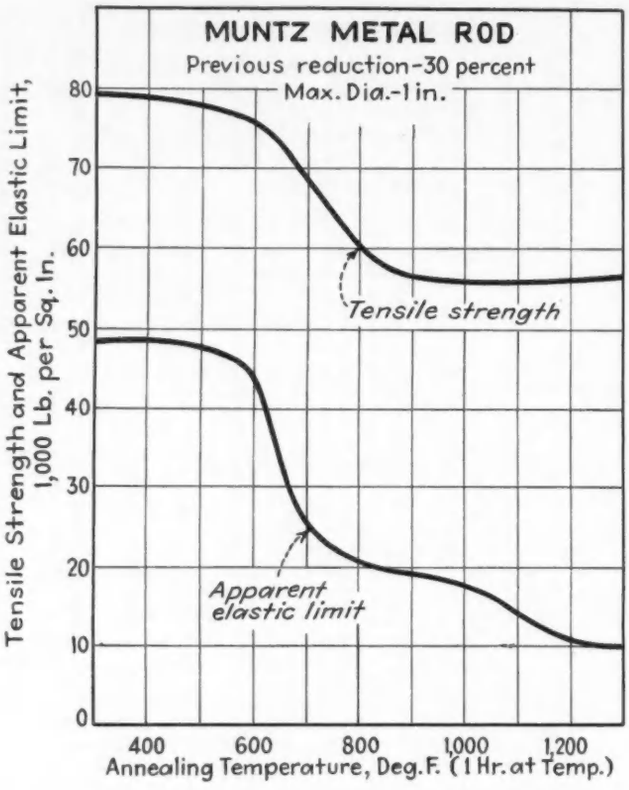

MUNTZ METAL ROD
Previous reduction-30 percent
Max. Dia.-1 in.
Tensile Strength and Apparent Elastic Limit, 1,000 Lb. per Sq. In.
Tensile strength
Apparent elastic limit
Annealing Temperature, Deg.F. (1 Hr. at Temp.)

Fig. 16—Effect of annealing on tensile strength and apparent elastic limit of rod previously cold drawn to 30 percent reduction of area from extruded material having a grain size of 0.010 mm.

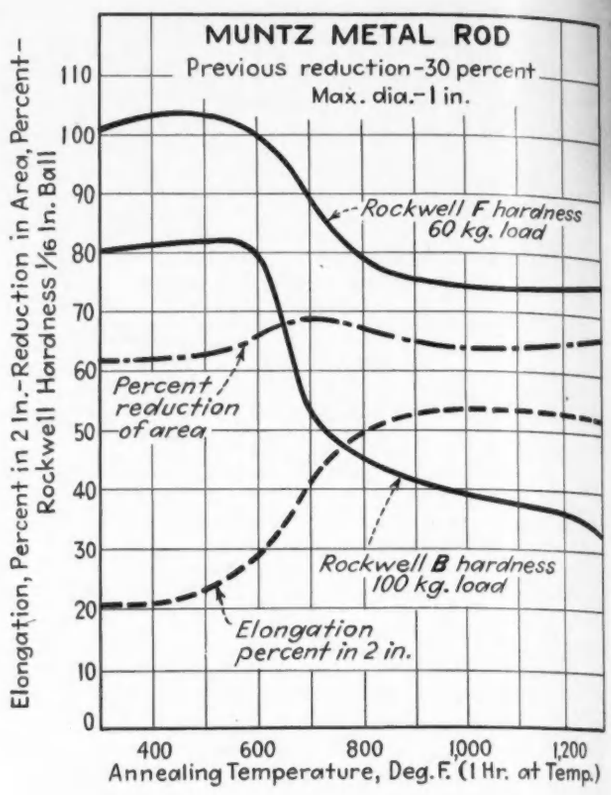

MUNTZ METAL ROD
Previous reduction-30 percent
Max. dia.-1 in.
Elongation, Percent in 2 In.-Reduction in Area, Percent-Rockwell Hardness 1/16 In. Ball
Rockwell F hardness 60 kg. load
Percent reduction of area
Rockwell B hardness 100 kg. load
Elongation percent in 2 in.
Annealing Temperature, Deg.F. (1 Hr. at Temp.)

Fig. 17—Effect of annealing on Rockwell hardness, elongation, and reduction of area of rod previously cold drawn to 30 percent reduction of area from extruded material having a grain size of 0.010 mm.

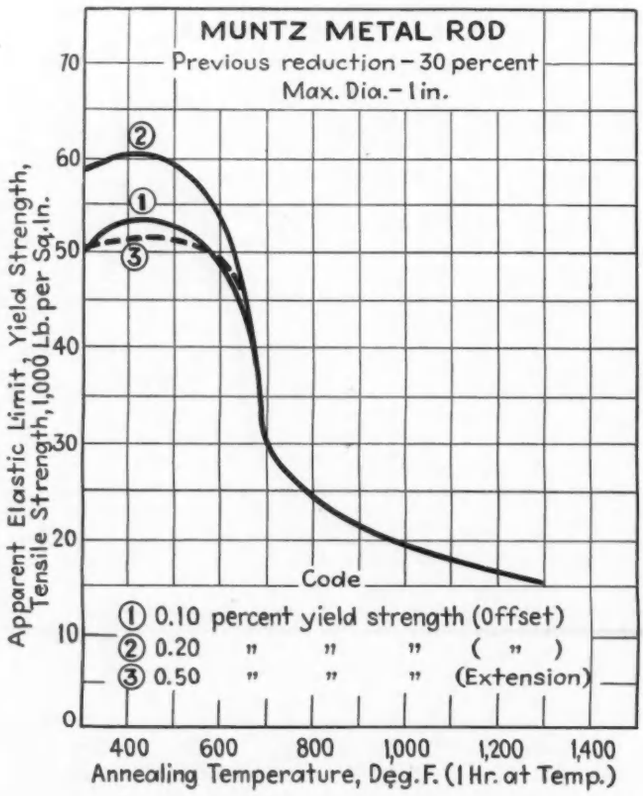

MUNTZ METAL ROD
Previous reduction - 30 percent
Max. Dia.- 1 in.
Apparent Elastic Limit, Yield Strength, Tensile Strength, 1,000 Lb. per Sq. In.
Code
① 0.10 percent yield strength (Offset)
② 0.20 " " " (")
③ 0.50 " " " (Extension)
Annealing Temperature, Deg. F. (1 Hr. at Temp.)

Fig. 18—Effect of annealing on yield strength of rod previously cold drawn to 30 percent reduction of area from extruded material having a grain size of 0.010 mm.

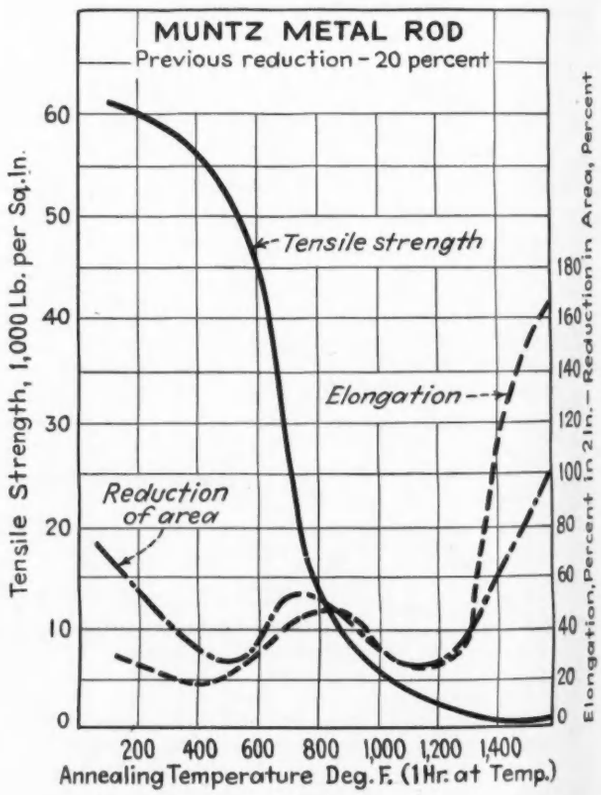

MUNTZ METAL ROD
Previous reduction - 20 percent
Tensile Strength, 1,000 Lb. per Sq. In.
Tensile strength
Elongation
Reduction of area
Elongation, Percent in 2 In. - Reduction in Area, Percent
Annealing Temperature Deg. F. (1 Hr. at Temp.)

Fig. 19—Effect of eleated temperature on tensile strength, reduction of area, and elongation of Muntz metal rod (62.40 percent copper) previously cold drawn 20 percent

Question and Comment

Author Amends Article on "Difference Calculus"

To the Editor:

I wish to thank Mr. Bruckner for his letter, published in March, drawing attention to the serious error in Example 1 and for his other comments on my article on "Difference Calculus."

Mr. Bruckner's points are very well taken and I should like to comment upon them in the order given.

1. The symbol list should have included the slope term S since it was included in Table II.

2. As he clearly shows, the I term should have been applied to the moment function before the first integration was performed. The accompanying table gives the correct solution for Example 1.

3. For applications where it is necessary to know the slope at points along the beam the constants would have to be left in their normal positions. If only the deflections are required, it would appear simpler to gather them together and apply them at one time.

4. The use of effective, in place of actual shaft lengths, would give more accurate results, but for average conditions (such as Example 2) would have only slight effect on the deflection curve.

—A. H. CHURCH
New York, N. Y.

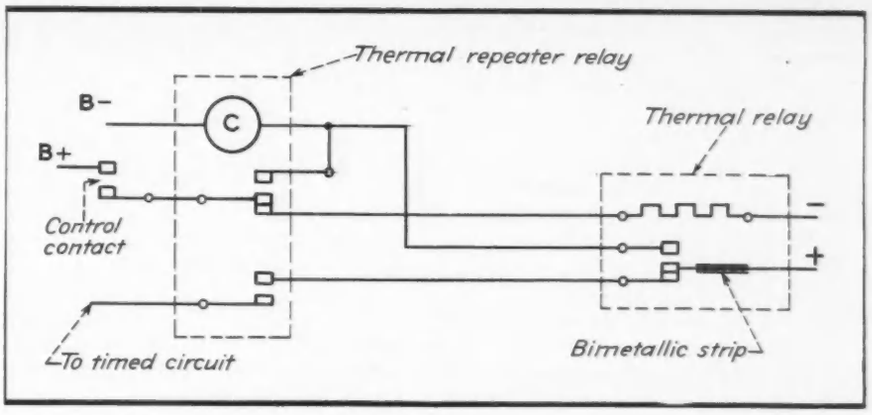
Table I—Solution for Beam Deflections (Revised)

X	t	b	$X\Delta X$	$\frac{X\Delta X}{bI^2}$	$\Sigma\frac{X\Delta X}{bI^2}$	$\Sigma\frac{X\Delta X}{bI^2}$	$\Delta X\Sigma\frac{X\Delta X}{bI^2}$	$\Sigma\Delta X\Sigma\frac{X\Delta X}{bI^2}$	y
0	4.0	2.0			0.3354			3.3120	0.00265
	4.0	2.4	2	0.0130		0.3289	0.6578		
2					0.3224			2.6542	0.00212
	4.0	3.2	6	0.0293		0.3078	0.6156		
4					0.2931			2.0386	0.00163
	4.0	4.0	10	0.0390		0.2736	0.5472		
6					0.2541			1.4914	0.00119
	4.0	4.8	14	0.0456		0.2313	0.4626		
8					0.2085			1.0288	0.00082
	4.0	5.6	18	0.0502		0.1831	0.3668		
10					0.1583			0.6620	0.00053
	4.2	6.4	22	0.0461		0.1351	0.2702		
12					0.1119			0.3918	0.00031
	4.6	7.2	26	0.0371		0.0933	0.1866		
14					0.0748			0.2052	0.00016
	5.0	8.0	30	0.0300		0.0598	0.1196		
16					0.0448			0.0856	0.00007
	5.4	8.8	34	0.0245		0.0326	0.0652		
18					0.0203			0.0201	0.00002
	5.8	9.6	38	0.0203		0.0102	0.0204		
20	6.0	10.0			0			0	0

Time Delay Relays

To the Editor:

In the February issue of PRODUCT ENGINEERING there is a slight inaccuracy that I wish to call to your attention. In Mr. Immels' article on "Time Delay Relays" he states that the thermals, items 7 and 12, are, at best, only fair in uniformity of time period. This statment may be considered inaccurate if thermal relays as a group are under consideration. There are thousands of thermal relays in use on American railroads to perform functions of vital importance when the time of operation must be within the meaning of the term "exact." There, relays operate under conditions of ambient temperature varying from —40 deg. F. of colder to +150 deg. F. or hotter and battery voltage variation from over-charge to condition of failure. The time of operation under these variables must be extremely close in order to avoid failure, or, in case of shortening of time, safety itself would be jeopardized. The circuit operation in general use is as shown. Note that the cooling period as well as the heating period is utilized.



The time range covers a range at least up to 10 minutes, with 3 to 6 the most commonly used. Note also from the circuit that the reset is practically instantaneous, dependent only on the release time of the thermal repeater. [As the writer is a government employee, his name is withheld.—EDITOR.]

To the Editor:

The description and data on thermal relays, items 7 and 12, in the article "Time Delay Relays and Their Characteristics" are only for thermal devices used in connection with industrial control. As far as exactness of operation is concerned, almost any device can be made to operate proportionately to its cost. As is well known, thermostats which employ a bimetallic strip are very accurate in their operation.

The most undesirable characteristic of a thermal relay is its slow make and break contact action unless a quick

acting mechanism is used in conjunction with the bimetal.

Some a.c. contactor magnet coils draw from 50 to 100 amp. with the armature in the open position. Obviously, the thermal relay contact pressure must be sufficient to prevent the contact from burning.

On d.c. coil circuits, the opening of the thermal relay contacts will not ordinarily open a very inductive circuit. Such a condition would burn the contacts as the arc would not be extinguished immediately.

I still rate this particular type of thermal relay as "poor" compared to the other devices. The electronic type which is rated "exact" will repeat to within 1/100 to 1/150 of a second.

Another inherent fault of the thermal relay is that its minimum period is too long for same applications. The thermal capacity and rate at which heat can be applied to the bimetal determines this minimum period.

—R. B. IMMEL
Ind. Control Engineering Dept.
Westinghouse Electric & Mfg. Co.

Laboratory Is Looking For Panel Mounting Valves

To the Editor:

We have been trying to obtain panel mounting valves for our laboratory here and have had difficulty locating a source of supply. Do you have a list of manufacturers who are making panel mounting valves of the "plug-cock" type in sizes from ⅛ to ¼ in. pipe size? We should appreciate any information you may supply us on this problem.

NORMAN C. PENFOLD
Armour Research Foundation

Can You Work This One?

H. E. SMITH

Solution to March problem—

Broke But Happy

Our customer who spent $10 in each of three stores after receiving from each proprietor as much money as he walked in with, had to begin his shopping tour with $8.75 to end up with nothing.

Proof: He must have entered the third store with $5 to receive another $5 from the proprietor and then spend exactly $10. To have had that $5 left after spending $10 in the second store he must have entered it with half of $15 or $7.50. To have had $7.50 left after the first store he most have entered with half of $17.50 or $8.75.

This month's problem—

Profit and Loss

A man sold two machines for $1,500 each. On one he had a profit of 25 percent. On the other he had a loss of 25 percent. What is his total profit or loss on the whole transaction? What percent loss could he afford on the second machine and still break even?

Case Histories in Patent Law

H. A. TOULMIN, JR.

NO. 28

QUESTION: How should invention contracts be handled to make them uniform and fair to everyone?

ANSWER: —The perennial problem of manufacturers, large and small, is how to handle the subject of the ownership of inventions and fair compensation for them. The mistake is usually made of confusing these two things. They must be kept absolutely apart and handled separately.

In the first place, the customary and fair way is for everyone working for the company, from the President down to the humblest mechanic, to sign a simple agreement to turn over to the company all inventions made in its line, whether in the time of the company or not, and whether at its expense or not.

The best way to handle this is to provide on the application for employment, a contract to this effect which is automatically signed at the time the applicant applies for the job. Then the person signing the agreement, as a part of his employment, understands full well the conditions of his employment.

On the other hand, special contracts are usually necessary for the person hired for engineering, research and development work. Different types of contracts are necessary for executives who are only collaterally interested in inventive work but from whom some of the best inventions usually come.

As a matter of fact, it has been the writer's experience that between eighty and ninety per cent of the best inventions have come from the top executives and not from the very fine engineering departments which their companies maintain. Perhaps this is due to a certain perspective, a more creative type of mind and balanced point of view which eyes the commercial necessities of providing a company with new products.

Engineers as a class are not inventive. Our industrial efficiency in creating new products would be greatly enhanced if we taught young engineers to use their heads to think with, rather than their fingers to play with slide rules. Mr. Charles F. Kettering of General Motors used to say: "A slide rule is a wooden substitute for brains."

The underlying rule which should be enforced by executives and which should be the basis for a policy established by a Board of Directors is this: Let the executive or employee do the fair thing of turning over the inventions in the company's line to the company, but they should set up a separate system for special rewards for special work of this sort in proportion to its value. This is the second half of the problem—The stimulation of invention and its reward is to establish fair rules applying to everyone in the company.

One steel company has the rule that anyone who invents while in its employ must give the idea to the company exclusively in its field, but the inventor is perfectly free to sell the invention or license it to non-competitors.

Another company has a Committee for periodically reviewing the results from employees' inventions and compensating the inventor accordingly.

Another company works the group plan by which groups who produce inventions, where it is difficult to see where invention starts and research ends, produce advantageous ideas for the company. In such cases the whole group is awarded some compensation.

In other companies the reward is left to the judgment of the management without any yardstick basis.

No matter what the plan adopted may be, my experience has been that a well run plan of compensation, fairly administered and honestly carried out, is one of the best guarantees of a loyal body of men and women who are doing their best for the company. It is an excellent insurance policy that the new things in the business will go to the benefit of the company.

News and Summaries

Mellon Researches Expanded

THE INDUSTRIAL RESEARCH STAFF of the Mellon Institute of Industrial Research at the University of Pittsburgh now consists of 205 fellows and 150 fellowship assistants. Nine new industrial fellowships started on their researches during the year, as follows: Calcite Technology, Coke-Plant Construction Materials, Fur, Lock Nut Technology, Naphthalene Chemistry, Packaging, Pine Chemicals, Tape Technology, and Tar Constitnents. In all, there have been 95 industrial fellowships in operation. Exigencies of the emergency have caused a considerable expansion in the activities of certain fellowships, particularly those concerned with coke-byproducts. The result has been many special studies for Federal Government agencies by various fellowships, such as those pertaining to heat insulation, petroleum, plate glass, protective coatings, synthetic organic chemicals and tar synthetics. In bringing out the particulars in his annual report, the Director, Dr. E. R. Weidlein said that the range of the programs of most fellowships has regard for postwar economic problems.

The report indicated special interest in the following continued industrial fellowship researches.

Anthrafloss, a high-grade mineral wool for insulating purposes, was made from anthracite byproducts by the multiple fellowship of Anthracite Industries, Inc. The multiple fellowship on heat insulation and roofing has aided in the development of an efficient insulating material suitable for temperature up to and including 1,500 deg. F. From the research program of the multiple fellowship on plastic metals has come the construction of a large commercial sponge-iron plant. Activities of other fellowships has benefited gray cast iron and steel production. Research of another fellowship has been on the refining of chromium. The multiple fellowship of the American Iron and Steel Institute has pursued steadily the utilization of waste pickling liquor. Investigational work includes the recovery of free sulphuric acid and copperas, the manufacture of iron carbonate and ammonium sulphate, the production of magnesium from low-grade ores by leaching with waste pickling liquor, and the preparation of sodium sulphate and ferric chloride.

In the field of ceramics, cast iron enameling, hollow-ware enameling materials, flat glass, and waste grinding sand from plate glass manufacture are projects of fellowships. Plate glass technology has brought about substantial advances in methods for evaluating flatness and texture of surfaces. Garspar, Garcolite, and Gartex are now made commercially from waste grinding sand as the result of the research and development work of the Mineral Products Fellowship. New markets have been made for Garspar in the fields of pottery and glass; Garcolite has been used as a pulverulent aggregate in concrete; Gartex is a processed filler for rubber, plastics, and paint.

Principal line of study of the multiple industrial fellowship on sulphur has been devoted to sulphur cement, to the reaction of gaseous olefins with sulphur, and to the employment of sulphur as a filtering medium. For removing sediment from water, sulphur was found to have an efficiency equal to or better than that of sand filters.

The multiple fellowship on gas byproducts has made new inorganic and organic thiocyanates, and a novel series of rubber accelerators has been discovered by this group. Other fellowships are investigating the upgrading of the components of coke oven light oil, the physical properties of tars and pitches and their improvements, new products from tars, and the problems of waste disposals in the by-product coking industry. A process has been devised for the commercial recoveries of cyclopentadiene, of great interest as a potential copolymer in the preparation of synthetic rubber and resins. Several types of natural gasoline and casing-head gasoline have been shown to contain amounts of isobutane and isopentane which make them desirable for sources of alkalation materials and for the production of 100 octane grade aviation fuel.

Additional fellowships are carrying on such projects as continued work on petroleum products, organic synthesis and development of new products, and investigations on the manufacture and applications of amines, detergents, lubricants, hydraulic fluids, and synthetic resins.

Mellon Institute is also carrying on research in food and nutrition, chemotherapeutic and bacteriologic researches, largely pertaining to the effects of the sulfonamides, and industrial hygiene.

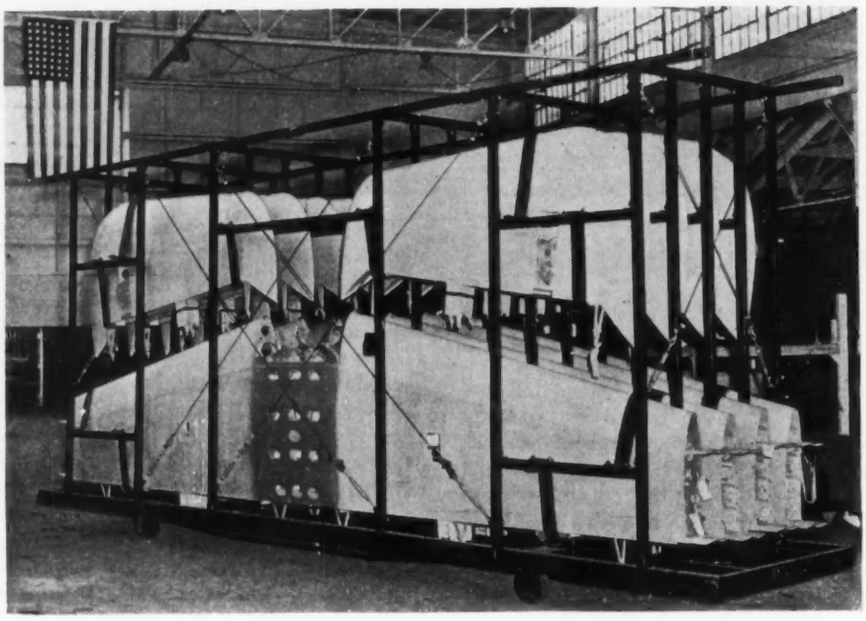
The Industrial Hygiene Foundation, a combined research and service association for health protection in the industries, maintain its headquarters at Mellon Institute. Investigations of this foundation have pertained to the effect of arc welding fumes on susceptibility to tuberculosis, to the reaction of living tissues, to silica granules, to the appraisal of X-ray films for use in physical examinations in the industries, and to the evaluation of present exhaust ventilation for the control of industrial health hazards. The foundation has been studying sick absenteeism and fatigue. Their report includes a review of the work on aluminum powder as a possible preventive in silicosis.

The fellowship on bone products has announced methods for obtaining higher use and grades of glue. This fellowship has lately been studying granular absorbents for use in sugar refining. Increases in the staff of the fellowship on rubber technology are being made for fundamental studies of underlying problems of the industry. Raolin, a special chlorinated rubber, the creation of another fellowship, is being allocated according to Federal instructions.

Of interest in connection with needs for plastic is the program of the multiple fellowship engaged in research on polymerizable silicon compounds. Fellowship research is also being conducted on the employment of plastics in water and gas meters and for many other purposes.

Casting Replaced by Pressed Metal

FROM C. A. Woodruff, chief of the Conversion Unit, Branch of Contract Distribution, W.P.B., comes the story of a manufacturer of parts for ladies' handbags who wanted to convert his capacity to war purposes. Up to recently he had turned out a million pieces of frames and fittings for handbags from sheet metal by cold presses. Looking over some of the samples of products



Bomber parts are carefully packed in special shock-absorbing shipping containers for transportation from the Fisher Body plant to another plant for final assembly

required by the government for the war effort, he chose a charger for a rapid fire gun as something he could handle. This charger was primarily a casting, but since the manufacturers had no machinery for such fabrications, he made a sample out of sheet metal, using only his existing machinery. (Note a similar application on page 242 of the April issue, PRODUCT ENGINEERING). The pressed device not only passed all the rigorous tests to which it was subjected before approval, but also was considered an improvement over its cast predecessor. Furthermore, it weighed only half as much.

Fisher Body Ships Plane Parts in Steel Containers

AS AN OUTGROWTH of their shipping system for automobile body parts, such as roof, door panels, and floors, Fisher Body is now shipping aluminum aircraft parts such as wings, rudders, stabilizers and ailerons, in steel frame containers. Each part is held in place in the container with strong but elastic straps which tend to absorb impact incurred during transport. Containers are equipped with rubber mounted knee action casters to absorb vibration while containers are rolled in or out of freight cars.

Several containers are shipped in each car, depending upon their length, and cars for this purpose are equipped with special brackets which grasp the containers at each end, while a horizontal jack between units keep them securely locked within the brackets. The jack provides protection against either forward or backward, or up and down motion, and trucks in which casters are locked prevent side sway.

Plane parts were formerly transported in wooden crates. It was found, however, that for such mass production as is now required, wood crates required excessive frame construction work, heavy charges because of the greater weight, and a complete loss of the packing material at the destination since it was impractical to return crates for repacking. Shipment by wooden crates required from two to three pounds of dunnage for each pound of goods. With the new steel containers, the ratio is about 1.5 lb. of dunnage to 1 lb. of goods. Furthermore, since the steel containers may be re-used many times, there is no loss of dunnage after each shipment, and it has been figured that for shipment of all bomber parts called for in the existing Fisher Body contract, only 1 oz. of steel will have been used as dunnage for each 1 lb. of aircraft parts shipped.

Meetings

National Electrical Manufacturers Association — The Homestead, Hot Springs, Va., May 10–15.

American Gear Manufacturers Association—Annual Meeting, Hershey Hotel, Hershey, Pa., May 11–13.

American Iron and Steel Institute—Annual Meeting, Waldorf-Astoria Hotel, New York, N. Y., May 21.

Society of Automotive Engineers—Annual Summer Meeting, The Greenbrier, White Sulphur Springs, W. Va., May 31–June 5.

American Society for Testing Materials—Annual Meeting, Chalfonte-Haddon Hall, Atlantic City, N. J., June 22–26.

Silver in Alloys For Aircraft Application

IN A RECENT REPORT issued by the American Silver Producers Research Project, the possibility was indicated that important use of silver will result from the application of magnesium alloys in the aviation field. Investigation on a recent magnesium alloy seems to bear this out. It contained 2.6 percent each of aluminum and zinc, 5.5 percent silver, and 0.2 percent manganese and had a specific gravity of only 1.89. In extruded bars, rolled sections, and large forgings, it is reported to have the following physical properties after suitable heat-treatment; tensile strength, 40,000 to 65,000 lb. per sq.in.; yield strength, 32,000 to 50,000 lb. per sq.in.; elongation, 2 to 10 percent; Brinell hardness, 80 to 95. Corrosion resistance is said to be comparable to all except the most corrosion resistant of commercial magnesium alloys. Although research on this type of alloy has not gone far enough to make further predictions, properties attained seem to indicate possibilities well worth more extensive investigation.

To Salvage 4,000,000 Pounds of Waste Paper

SALVAGE OPERATIONS at the Glenn L. Martin Company will include the saving of 4,000,000 lb. of waste paper which formerly was burned. Since the waste paper of this concern contains a large amount of valuable information, each day's collection was reduced to ashes before the eyes of armed guards. In the new system, paper is shredded and baled so that it can be turned back to use.

Do You Know That—

A modern battleship requires 30 tons of blueprints before construction is even started; 5,000 different blueprints must be drawn for a combat plane, 2,500 for a tank, 250 for a rifle, and 40 for a simple looking object such as a shell. (6)

A NEW PLASTIC SHEET MATERIAL has the appearance of heavy metal foil of gold or silver finish. (7)

ENGINEERING IN WASHINGTON

McGRAW-HILL WASHINGTON BUREAU, *PAUL WOOTON, CHIEF*

Steel and wood replacing a large percentage of dural in airplanes for the Army seems to be one of the important materials substitution in this war so far. North American Aviation announces a shift in its AT-6 trainer to low carbon low alloy steel, and some wood, reducing aluminum poundage by 75 percent. And Curtiss-Wright has made a deal with the Army to supply nearly-all-wood transport planes.

First question asked by most persons on hearing this news is: "Why hasn't steel been substituted for aluminum in airplanes before now?" The answers are not very satisfactory. Fleetwings, Inc., built a successful plane of stainless steel, but that steel contains a high percentage of strategic alloy and, like aluminum, is scarce. Manufacturers didn't have the right steel; N.A.A. has been working on a cold rolling process for over a year. Some of the best engineers say that designers simply got into the aluminum habit and didn't like to explore other materials.

Prettiest points about steel planes: The skin can be spotwelded, giving a surface as smooth as glass and having far less drag in the air than riveted dural; pure aluminum can be welded, the engineers say, but dural, the aluminum alloy of which planes are made, is hard to handle; the N.A.A. steel can be produced in plentiful quantities. It's not a very important point, but dural weakens at medium-high temperatures, such as in ordinary fires, while steel does much better.

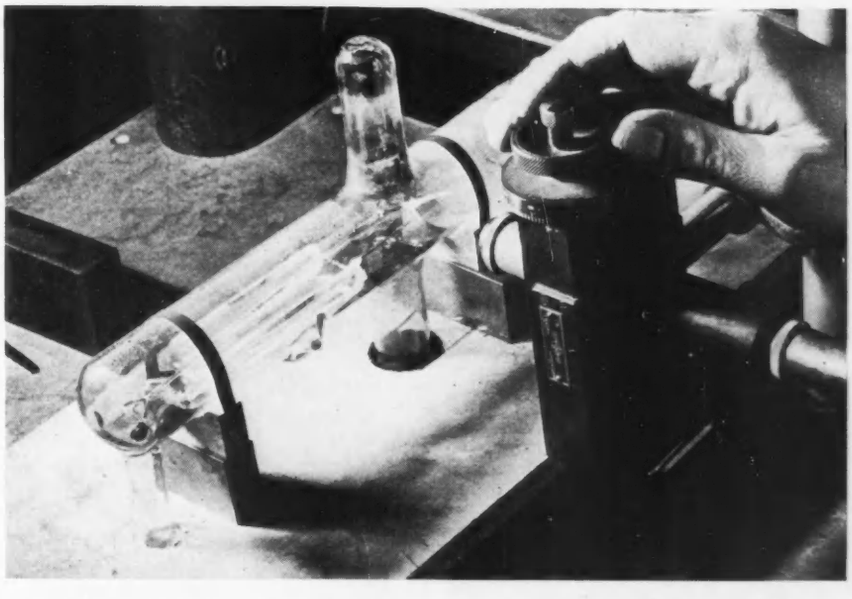
For each 4,000-lb. trainer, 1,246 lb. of aluminum alloy will be saved for other war production. One thousand such trainers would save 623 tons or enough for 150 medium bombers. The steel and wood AT-6 will be about 150 lb. heavier than the dural model, but the AT-6 permissible design overload is 350 lb., and designers believe they can put the steel model down to dural weight in a little while.

Curtiss-Wright's wood transport, designed by George A. Page, will be built in a new plant, the company's fifth. The transport will have the range and speed required by modern warfare. A large proportion of the parts will be subcontracted by furniture and other woodworking companies. This will bring as-yet-non-defense equipment and manpower into the picture. Experts from early wood design projects, like Fokker, are being brought back into service.

Conversion to strip steel, by merchant shipyards, instead of sheared plate, has speeded work on EC-2 (Liberty) ships, now coming at the rate of nearly one a day. Mills that had made strip sheet for automobiles were easily converted by opening the roller gap for thicker plate. The change in the yards to strip would have been quite serious if seams were still riveted. But they are all welded now, and it's easy to turn a farm boy into a welder in a few weeks.

Steel plate users are being asked by WPB to make all the use they can of plate less than ¾ in. thick by 72 in. wide. About one third of the potential plate making capacity of the country is in the conversion of strip mills, most of which can make only the thinner and narrower sizes. The output of the sheared plate mills, about one quarter of the total, is directed to heavy war jobs that cannot use strip.

Canister shot is the new standard with Ordnance; shrapnel is out, and no more will be manufactured. Shrapnel is a shell, or projectile, loaded with a base charge of explosive and a quantity of ball. It explodes over the target at an elevation determined by the setting of the fuse that detonates the base charge. The ball is fired from the shell, which is really a sort of traveling shotgun. Canister, as the word suggests, is a can full of ball—specifically 122 of them, ⅜ in. in diameter. The can is fired from a gun, usually a field piece. It contains no explosive. As soon as it is out of the muzzle, the can starts to disintegrate, and the flight of ball continues, exactly like a charge of shot from a shotgun. Shrapnel was for use against troops in trench warfare. But now the troops are moving rapidly, many in tanks and behind other armor too thick for shrapnel. Canister is used mostly against machine gun nests.

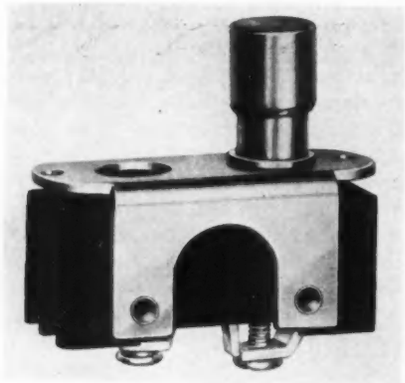


Weighing corrosion or tarnish on steel or other metals, this instrument, developed by Dr. Earl A. Gulbransen, Westinghouse Research Engineer, measures the weight of a single layer of oxygen atoms. The white glass weighing instrument inside the tube, left, is measuring in units of ten billionths of an inch the weight of corrosion on a steel sample half the size of a razor blade. An invisible strand of tungsten wire suspends the weighing beam. Below the table, inside the vertical tube, the piece of steel is hung by fine wire from the right-hand end of the beam.

New Materials and Parts

Aircraft Switch

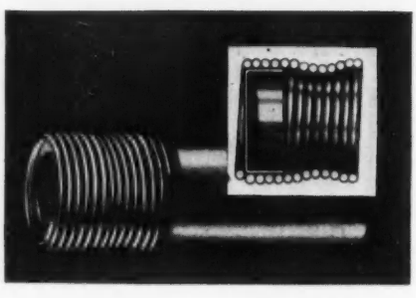
Designed to meet the requirements of United States Air Corps specification 94-32249 for aircraft applications. The over-plunger mechanism, mounted on a steel bracket, has a pre-travel of 1/16 in. minimum and an overtravel of 1/8 in., plus or minus 3/32 in. Pressure required to operate the switch is 1 lb. minimum and 8 lb. maximum over the entire operating range. Ratings are 25 amp. at 28 volts d.c., 17 amp. at 125



volts a.c. and 8.5 amp. at 250 volts a.c. non-inductive loads. Can be supplied normally open or normally closed. Special mounting lugs are provided on the bottom of the switch to allow the use of ring type wire connectors. The Mu-Switch Corp., Canton, Mass.

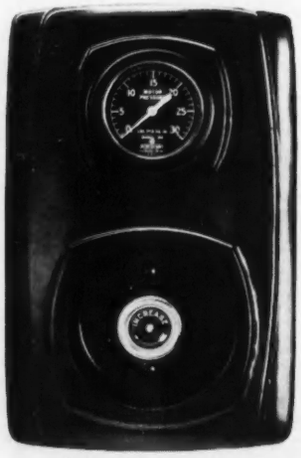
Protector for Tubing Ends

Flexible metal caps designed to protect plain end and beaded tubing from handling knocks and shocks and to seal out dirt, dust, and moisture. Called Flex-caps, they consist of a steel spring of hour-glass design. Turning this spring in a counter clockwise directtion expands it to accommodate the tubing. When released, the spring seizes the tubing. These caps are available to fit all standard tubes from 3/16 to 2 in. Tubing Seal-Cap, Inc., 215 W. 7th St., Los Angeles, Calif.



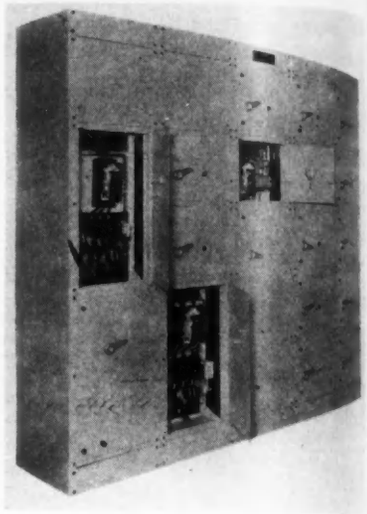
Remote Valve Control Unit

Practical for operation over distances as great as 1,000 ft., this unit has been designed for the manual control of remote valves, dampers, and pneumatically operated equipment. Flush mounted, its front surface is 1/16 in. from the surface of the instrument panel. Possible danger from accidental bumps is eliminated through the recessing of the setting knob and pressure indicator. Unit may also be used for the setting of control points of different instruments or the adjustment of positioning and pressure producing pistons located at inaccessible points. The Foxboro Co., Foxboro, Mass.



Motor Control Centers

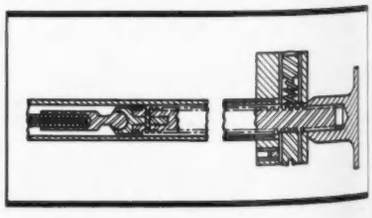
Group control of a-c motors up to 100 hp. is the function of a new control center with interchangeable units. Obtainable in 76- and 90-in. heights, centers consist of a 7-in. filler at the top for horizontal buses and cable troughs, a 5-in. filler at the bottom for cross cable duct, an 8-in. filler or master terminal board, and multiples of the basic 14-in. starter unit heights. Interchangeable units are trim mounted 20 in. wide, permitting two to five control elements to be placed in a single section. Sections 20 in. deep allow the mounting of units



on each face, back to back, if desired. For operation on 3-phase, 220, 440 and 550 volt circuits, the motor starter unit consists of a full voltage magnetic starter providing external manual reset and thermal overload and low voltage protection. Disconnect switches may be unfused, quick-make and quick-break, of the circuit breaker type with thermal inverse time overload and instantaneous short circuit protection. Fused disconnect switch is available up to 7.5 hp., 220 volts; 15 hp., 440 volts. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Remote Control Vernier Head

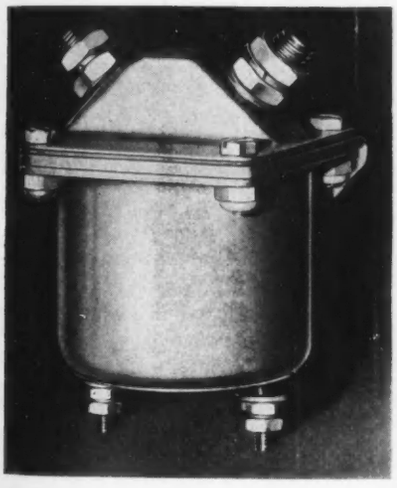
Combining a positive lock, vernier adjustment, and push-pull movement, this remote control head can be mounted on any type control, with a sliding flexible member of piano wire or wound spring designed to take a tension load. The small pin set at the top of the base plate rides in the groove of a thread when the pressure exerted by the spring in the bottom of the base forces the plunger, or movable threaded rod, into engagement. This engage-



ment is released when the pressure of the hand on top of the knob depresses the spring, and allows the plunger to move freely. Upon release of hand pressure, the spring forces re-engagement making a positive lock. Arens Controls, Inc., 2253 So. Halsted St., Chicago, Ill.

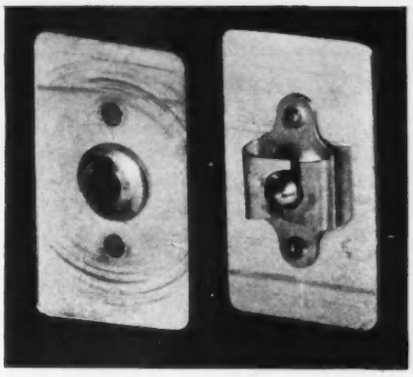
Dynamotor Contactor

Especially designed to start and stop dynamotors used with aircraft equipment, also tank installations. Copper-lead-alloy contact tips are used to assure good high-current inrush performance. Meeting Signal Corps vibration requirements, the contactor is good for mechanical frequencies of 5 to 55 cycles per sec. at a maximum of $\frac{1}{32}$-in. amplitude ($\frac{1}{16}$ in. total travel), applied in any direction. Single-pole, normally open contacts are designed to stay open when the coil is not energized and closed when the coil is energized even when the contactor is subjected to a linear acceleration of ten times gravity, applied in any direction. The contactor is suitable for use in ambient temperatures ranging from 60 deg. C. to minus 40 deg. C. Contact rating is 50 amp. on an 8-hour basis. Inrush current rating is 500 amp. at 32 volts d.c. Coil wattage is 9.5. Available for either 12 or 24 volts d.c. the dynamotor contactor measures approximately 2½ x 4 in., weighs only 2.3 lb. General Electric Co., Schenectady, N. Y.



Fastener for Metal Sheets

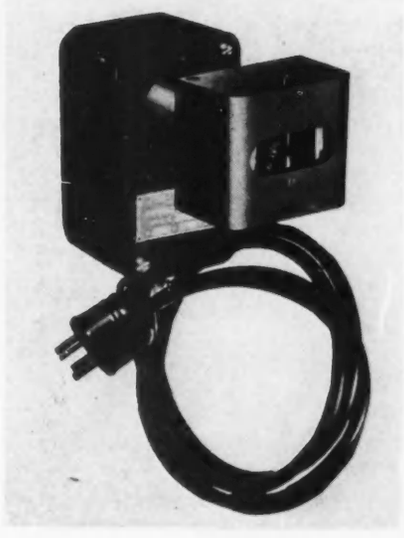
This fastener is made up of two primary parts, a stud and a stud receptacle. It has been developed for use on aircraft engine cowlings, hand-hold covers, and other aircraft applications. Features claimed for the device are great strength in tension, minimum deflection between sheets, high shear



strength, vibration resistance, and ease of installation. Members to be joined need not have close tolerances, and the load does not come on the spring part. Known as the "Airloc," it can be used on any combination of thicknesses. United-Carr Fastener Corp., 29 Ames St., Cambridge, Mass.

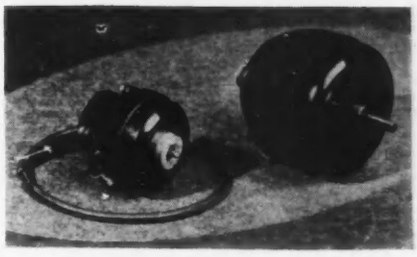
Light Scanner

Consists of a light source and photo tubes in a single compact housing which may be attached to any sensitive photo-electric amplifying system. It can be used wherever photoelectric control is desired from a reflecting surface two inches or less distant from the unit. Registration marks on moving materials, or the travel of a meter pointer into the spot of light, causes a change in the illumination reaching photo tubes. Responses of the photo tubes are transmitted through co-axial cable to a remote amplifier, making possible the control of auxiliary electric devices. Other applications are: sensitive cut-off from contrasting colors on similar materials, control according to perforations punched or marked on automatic business machine cards, and in conjunction with mirrors, galvanometers and mirror compasses. Scanner is sufficiently flexible to operate with any standard sensitive amplifier, although an amplifier, Model 668, has been specifically designed for this service. United Cinephone Corp., Torrington, Conn.



Fan Motors

Unit-bearing, shaded-pole fan motors for applications requiring short, quiet compact units, have been developed especially for use with commercial refrigeration, air conditioning and ventilating systems. There are three models for 115-volt, 60-cycle operation. Oil in a large reservoir is transferred to the shaft through wool felt, and is then



pumped through the bearings and into the oil return, from which it flows back to the reservoir. End-bump and axial-thrust forces are counteracted by the magnetic forces tending to hold the rotor in alignment with the stator. This provides a frictionless, non-wearing thrust arrangement. A special mounting arrangement on some sizes permits the use of simple low-cost formed fan blades without hubs. General Electric Co., Schenectady, N. Y.

Phenolic Molding Compound

Particularly developed for radio parts such as condensers and crystal holders, a new phenolic molding compound is now available. It has a low power factor (0.07 at 1,000 kc) and high dielectric breakdown and fatigue. Designated as Durez 11863, it is said to have excellent moldability. It is neutral in color. Durez Plastics & Chemicals, Inc., North Tonawanda, N. Y.

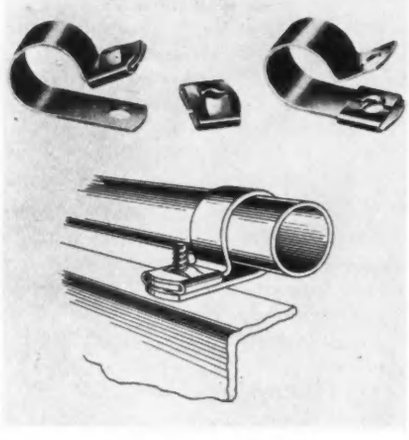
Insulation Tubing

"Fibronized Koroseal," a tubing designed for electrical insulations, is said to have high elasticity, inside and outside smoothness, close manufacturing tolerances, and excellent resistance to heat and many chemicals. Fire proof, it possesses an insulation resistance of infinity after 16 hours of 90 percent relative humidity and 105 deg. F. Flexibility is said to have been retained after

subjection to 225 deg. F. for approximately 1,000 hours, as determined by A.S.T.M. tests. It has tensile strength of 2,845 lb. per sq.in., dry dielectric strength (.022 in. wall thickness) of 1,050 volts per mil, a wet dielectric strength (.022 in. wall thickness) of 817 volts per mil after 24 hours emersion. Tubing is furnished in A.S.T.M. sizes and a variety of colors including a transparent shade. Irvington Varnish & Insulator Co., Irvington, N. J.

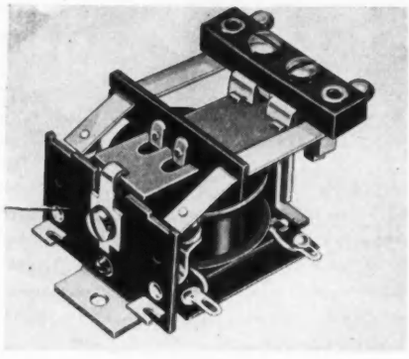
Conduit Clamps With Speed Nuts

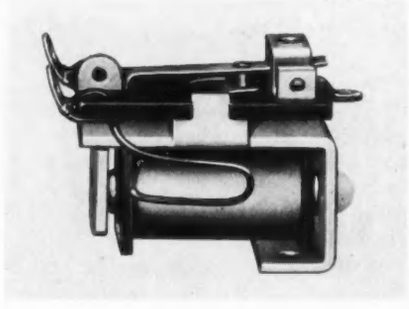
Standard AC-755 conduit clamps are now furnished with Speednuts attached to either the upper or lower leg for faster assembly. In addition to the savings in handling time, the new Speednut attached clamp is said to weigh less than half as much as a standard hex lock nut formerly used. Tinnerman Products, Inc., 2041 Fulton Rd., Cleveland, Ohio.



Relays

Two new relays recently introduced are the Series 165 Vibration Resistant Relay, and the Series 195 DC Midget Relays. According to the manufacturer, the exceptional resistance to vibration of the former is achieved by careful counterbalancing and compact design of contact assemblies and its generally sturdy construction. Relays are entirely insulated from ground and tested with a minimum of 1,500 volts. Field piece and armature are of annealed magnetic iron, and tinned phosphor bronze blades carry heavy silver points. Contact combinations up to double pole, double throw have contact capacities up to 12.5 amperes per pole, measured at 110 volts, 60 cycle, non inductive a.c. Contact rating for aircraft use is 8 amperes at 24 volts d.c. Other voltages and contact combinations are available. Overall dimensions are 2½ x 1 13/16 x 1⅝ in., weight (DPST Contacts) is 5 oz. Particularly for aircraft and radio work, the Midget relay has a maximum contact capacity of approximately 150 watts, measured at 110 volts, 60-cycle, non-inductive a.c. Maximum contact





combination available is double pole, double throw. Relay coil in this relay is available for any voltage up to 75 volts d.c., maximum resistance 3,000 ohms; average power requirements of the coil are 2.5 watts. A light armature operates the contacts which are mounted on an insulated (900 Hi-Pot test) from the field piece. Contact blades are tinned phosphor bronze with contact points of fine silver. Field piece and armature are annealed magnetic iron. Coil is varnish impregnated and baked. With single pole, double throw contacts, weight of the relay is .85 oz. Guardian Electric Mfg. Co., Chicago, Ill.

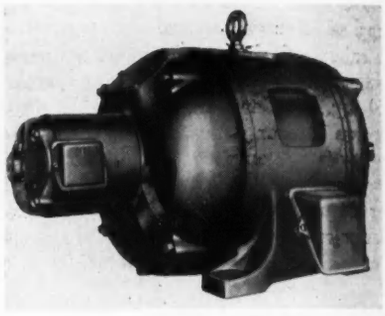
Motor Units for Aircraft

Units can be supplied with basic motor only, basic motor with clutch, basic motor with single or double gear reductions, and basic motor with single

or double gear reductions and magnetic clutch. Recommendations for applications include cowl and wing flap controls and oil cooler controls. Called the E2Y1PC, this motor can be obtained with ratings of 1/6 hp. at 1,500 r.p.m., 1/8 hp. at 1,500 r.p.m., 1/6 at 200 r.p.m., and 1/8 at 200 r.p.m. The Dumore Co., 14th and Racine St., Racine, Wis.

Revolving Field Generators

Streamlined, revolving field, alternating current generators are available in sizes from 7½ to 75 kva. These generators are wound for the standard voltages, single-phase 2 and 3 wire, 3-phase, 3 and 4 wire, or 2-phase, 4-wire. For light and power, the neutral may be brought out for 3-phase, 4-wire systems. They are built to meet A.I.E.E. and N.E.M.A. voltage regulation standards. An automatic voltage regulator is generally used for lighting service where



the load fluctuates through a wide range. They can be designed for any desired value of power factor, but the regulation will be covered by the power factor. Century Electric Co., 1806 Pine St., St. Louis, Mo.

Resistance Welding Timer

For use in automatic resistance spot, butt, or projection welding. Fully electronic, the timing circuit consists essentially of a resistor capacitor circuit whose charging rate is adjustable by varying a resistor. Voltage of the circuit is applied to the grid of a thyratron tube so that after a predetermining charging time has elapsed, the thyratron is energized and operates the relay. When welding thick plates, brass or stainless steel, timer permits intermittent heating and cooling steps of from 3 to 30 cycles. On the dead front panel are mounted the individual timing circuits, potentiometer knobs and dials, repeat and non-repeat switches, and the tubes. Control transformer is wound for 115, 230, 460, 515 volts, 50-60 cycles. Solenoid relay is rated 25 amp. at 110 or 220 volts, and 10 amp. at 440 or

550 volts. Except where 24 volt circuit operation is specified, standard foot switch control voltage is 115 volts. Unit is 17¼x12x12 in. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Balanced Hydraulic Valve

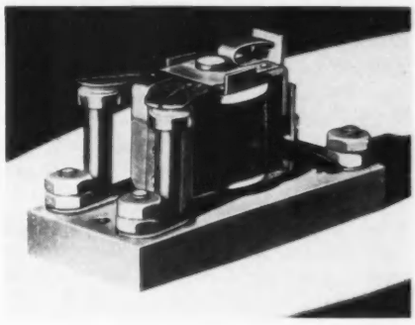
To eliminate "pressure-locking" at pressures of 1,000 lb. or more, a new type of high-pressure hydraulic operating valve has been developed. It is said that it is just as easy to operate from either "on" position or from "neutral." The need for maintenance has been reduced through a simplified design with only one moving part. The valve is built in four sizes, ⅜ in., ½ in., ¾ in. and 1 in. It is called the "Nopak Hy-Pressure Hydraulic Valve." The Galland-Henning Mfg. Co., Milwaukee, Wis.

Electrically Conductive Rubber Hose

A new sandblast hose which does not contain any internal wire construction for dissipating static charges. Resistance of old rubber compounds on conventional sandblast hose exceeded one quadrillion ohms per c.c. The resistance of the new compound is only 3,000 ohms per c.c. This hose not only carries away the high static charges, but also is resistant to the abrasive action of sharp sand particles. It is made in 4-ply construction, in sizes ranging from ¾ to 3 in. The B. F. Goodrich Co., Akron, Ohio.

Aircraft Relay

For operation at high altitudes under severe vibration conditions, a lightweight relay has been designed particularly for aircraft applications. Designated CR2791-A100A, the relay weighs 4½ oz., measures 3⅛x1⅝x1⅜ in. It can be mounted in any position. It is designed for mechanical frequencies of 5 to 55 cycles per sec. at $\frac{1}{32}$ in. maximum amplitude ($\frac{1}{16}$ in. total travel) in any direction. It may be used in altitudes from sea level to 14,000 ft., at ambient temperature ranging from minus 40



deg. C. to 93.5 deg. C. It has a current rating of 25 amp. at 12 or 24 volts. The coil operates at 1.2 watt. Single-pole, normally open contacts are designed to stay open when the coil is not energized, and closed when the coil is energized at rated voltage even when subjected to linear accelerations of ten times gravity in any direction. Built to meet United States Army Air Corps specifications, it is corrosion proof. General Electric Co., Schenectady, N. Y.

Machinable Ceramic Body

Ceramic bodies in the semi-plastic or unfired state can now be obtained for the development of sample or test pieces in a customer's own organization. After machining to desired size and shape, sample pieces are then returned to the manufacturer for proper firing and conversion into finished Crolite, unless the customer prefers to do his own firing. Blocks, rods, or tubes furnished for outside machining are usually of pure magnesium silicate, providing excellent electrical and mechanical characteristics in the final fired state. Henry L. Crowley & Co., Inc., West Orange, N. J.

Blackout Paint

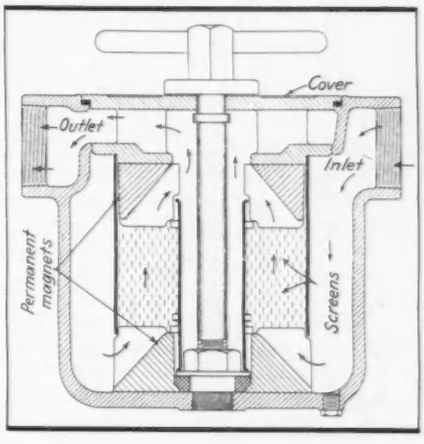
Waterproof paint which can be brushed or sprayed provides a dull finish blackout in one coat. Manufacturers claim that it will resist heat up to 210 deg. F. Coverage is approximately 500 ft. to the gal., depending upon the smoothness of the glass. It dries to touch in 15 min., is hard in two hours. Can be removed with kerosene and other mineral solvents. Hilo-Varnish Corp., 42-60 Stewart Ave., Brooklyn, N. Y.

High Strength Solder

Special solder developed for metals such as aluminum, aluminum alloys and Dow metal. Manufacturers claim that it is free-flowing and that it provides high tensile strength in sheet metals and extruded aluminum. It retains the color of aluminum and Dow metal and is corrosion resistant. Rolled ⅛ in. square, it runs approximately 270 in. to the pound. Lloyd S. Johnson Co., 2241 Indiana Ave., Chicago, Ill.

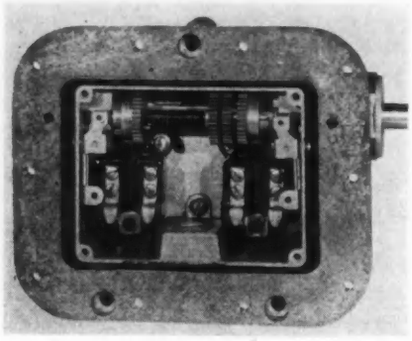
Filter for Lubricating Systems

To remove fine iron and steel particles from circulating coolant systems, a permanent magnet Ferrofilter, designated as the PQ-6, is said to have greater capacity than any previous Ferrofilter of this type. It has a capacity of 200 g.p.m., pipe-size connection of 3 in., a height of 15⅝ in., and no moving parts. S. G. Frantz Co., Inc., 161 Grand St., New York, N. Y.



Rotating Type Limit Switches

Two new forms of CR9441-C2 rotating-type limit switches are housed in heavy flanged, cast iron inclosures. One form is for class 1, group D hazardous gas location, and the other is for applications requiring a water-tight switch. Contacts operate between a minimum of ½ turn of the driving shaft and a maximum of 120 turns. Overtravel will not harm the switch mechanism. Double-break, fine-silver contacts clean themselves by opening and closing with a



rocking motion. New contact points can be installed with a screwdriver. Double-break contacts located at each end of the travel can be made normally closed or normally opened without additional parts. Conduit entrance of ¾ in. and a roomy box allow rapid and easy wiring. General Electric Co., Schenectady, N. Y.

Impact Resistance Plastic

New impact-resistance phenolic molding plastic. Bakelite phenolic resin XM-15000, developed to meet the need for a material that can be preformed on automatic tabletting machines. Molded properties are: Specific gravity, 1.36; weight per cu. in., 22.2 gm.; tensile strength, 5,000 to 5,800 lb. per sq. in.; modulus of elasticity, 10–13 x 10^5 lb. per sq. in.; impact strength, 4.8–6.3 ft. lb. per sq. in.; molding shrinkage, 0.005–0.007 in. per sq. in.; flexural strength, 8,400–8,800 lb. per sq. in. Recommended for use where molded parts are to be subjected to temperatures up to 300 deg. F. (149 deg. C.). To secure good electrical properties the material must be preheated. Bakelite Corporation, 30 E. 42nd St., New York, N. Y.

Dry Developing Printing Paper

Improved ammonia dry-developed printing paper is said to produce a deeper, sharper blue or red line print on a clean white background. It is further claimed that every roll, either in the same shipment or different shipments, is entirely uniform, both as to speed and color, so that printing and developing machines do not have to be set to conform with any variance. Its 50 percent rag content basic paper stock stands up under creasing and hard usage. Manufacturers offer a free five-yard sample of Vapopaper to engineers equipped to produce ammonia dry-developed prints. Specify preference for blue or red line, as well as regular or fast-printing speed. The Frederick Post Co., Box 803, Chicago, Ill.

Books and Bulletins

Plastics in Engineering

JOHN DELMONTE — *Second Edition. 601 pages, 6½x9¼ in. Blue imitation leather covers. Published by Penton Publishing Company, Cleveland, Ohio. Price $7.50.*

To keep his authoritative book apace with the fast-moving plastics industry, John Delmonte, director of the Plastics Industries Technical Institute, has incorporated into this second edition information on such new plastics as plyamides, polyvinylidene chloride, and methyl cellulose. Discussion of new materials, new techniques and results of data on physical, electrical, thermal and optical properties, as well as new information on plastic-bonded plywood, adhesives, extrusion and synthetic rubber and rubber-like resins make the book one of the most complete and up-to-date works available.

The first few of the 26 chapters of the book give a comprehensive description of the various types of plastics available. The next few chapters cover design and specification of plastics and cover mold design and molding techniques. Of specific interest to design engineers are several chapters on industrial applications, such as gears, cams, styling, etc. Data on fabricating and finishing are also included.

The Condensed Chemical Dictionary

THOMAS C. GREGORY, *Editor. Third edition, 756 pages, 6x9 in. Red buchram covers. Thumb index. Published by Reinhold Publishing Corp., 330 West 42nd St., New York, N. Y. Price $12.*

Over 6,000 new items have been added to this revised edition of a work already well established as an authoritative collection of data valuable to chemists and engineers. Popular names of chemicals or chemical materials, chemical terminology and trade or brand names of chemical specialties are included. A new feature is the incorporation of "Typical Specifications" which, furnished by manufacturers, describe products as marketed or sold commercially.

In style the dictionary approaches the encyclopedic form, with cross references, brief descriptions or derivations, chemical and physical properties, grades, containers, uses and fire hazards. All data, including that on labelling in accordance with Railroad Shipping Regulations, has been brought up-to-date.

The book is prefaced with a guide to the pronunciation of chemical names, and tables showing the effect of wartime on chemical prices. In the appendix are chemical tables, tables of equivalents, specific gravities, freezing points, boiling points, etc.

Gages and Their Use in Inspection

FRED H. COLVIN—*157 pages, 4¾ x 7½ in. Red buchram covers. Published by the McGraw-Hill Book Co., Inc., 330 West 42nd St., New York, N. Y. Price $1.50.*

Principles, applications and maintenance of industrial gages are adequately covered in a surprisingly short space of pages and reading time. The author's straightforward style and great familiarity with his subject helps make this a book that can be thoroughly digested, with profit, by engineers, inspectors and production men alike.

In addition to the mechanics of gaging, subjects covered include accepted terminology of shops, limits and tolerances, outlines of construction of gages of foremost types, interchangeable manufacture and inspection, machine tool inspection and types of fit.

• • •

Tests of Cylindrical Shells

University of Illinois Engineering Experiment Station Bulletin Series No. 331. Paper covers, 129 pages, 6x9 in. Published by the University of Illinois, Urbana, Illinois. Price $1.

A report of an investigation conducted by the Engineering Experiment Station, University of Illinois, in cooperation with the Chicago Bridge and Iron Company by Wilbur M. Wilson and Emery D. Olson, to determine the action of thin cylindrical steel shells when subjected to axial compression, transverse shear, and flexure, singly or in combinations. Tests were made on specimens machined from 4-in. seamless steel tubing, and on 30 in. O.D. shells welded of ¼-in. steel plates, except for one series having a 28-in. outside diameter. Following a report of the various tests, the bulletin concludes with an analysis of the stresses induced in cylindrical shells when loaded as beams.

TOLERANCES AND ALLOWANCES

Nomographic Determination

FITS AND TOLERANCES of holes and shafts or other mating parts, as computed from American tentative standard B4a — 1925 of the American Standards Association, can be determined quickly with the chart on the following page and a straight edge. Straight lines through the selected value on the nominal dimension scale at the left and two points in the center of the chart will indicate desired allowances and tolerances on the right-hand scale.

The 11 points in the center of the chart are numbered to correspond with the classes of fit for which they are used and are labeled to indicate the result or results to be obtained from each on the right-hand scale. Allowances for classes 4 and 5 are zero for all nominal dimensions of mating parts and, therefore, are not found on the chart. Only two solution lines or two positions of a straight edge are required for any one nominal dimension.

The right-hand scale is read only to the closest ten-thousandth, as indicated by the following examples which may be checked on the chart on the following page. Dimensions are given in the standard basic hole system.

RECOMMENDED APPLICATIONS OF CLASSES FOR INTERCHANGEABLE MANUFACTURE

Class 1: Loose fit, large allowance. Provides considerable freedom: used where accuracy is not essential. Employed on agricultural and mining machinery; controlling apparatus for marine work; textile, rubber, candy, and bread machinery; and general machinery of a similar grade.

Class 2: Free fit, liberal allowance. For running fits where speeds of 600 r.p.m. or over and general pressure of 600 lb. per sq.in. and over. Intended for use in electrical machinery, engines, many machine tool parts, and some automotive parts.

Class 3: Medium fit, medium allowance. For running fits under 600 r.p.m. and with general pressures less than 600 lb. per sq.in.; also for sliding fits, the more accurate machine tools, and automobile parts.

Class 4: Snug fits, zero allowance. Closest fit that can be assembled by hand and necessitates work of considerable precision. Used where no perceptible shake is permissible and where moving parts are not intended to move freely under load.

Class 5: Wringing fits, zero to negative allowance. Assembly is usually selective and not interchangeable.

RECOMMENDED APPLICATIONS OF CLASSES FOR SELECTIVE ASSEMBLY

Class 6: Tight fit, slight negative allowance. Light pressure required to assemble. Parts are more or less permanently assembled, such as fixed ends of studs, hook eyes, pulleys, rocker-arms, etc. Used for dry fits in thin sections or extremely long fits in other sections, also for shrink fits on very light sections.

Class 7: Medium force fit, negative allowance. Considerable pressure required to assemble, parts considered permanently assembled. Used in fastening locomotive wheels, car wheels, armatures of electrical machinery, and crank disks to their axles. Used for shrink fits on medium sections or long fits. These fits are tightest that are recommended for cast-iron holes or external members since they stretch cast iron to its elastic limit.

Class 8: Heavy force and shrink fits, considerable negative allowance. Used for steel holes where metal can be highly stressed without exceeding elastic limit.

[See "Fits and Tolerances—Examples of All Classes," by Dr. Ernesto Geiger, on page 266 of this number.]

Example 1: 3½-in. shaft and hole, Class 2 fit. (see Fig. 1)

Shaft and hole tolerance = 0.0020 in.

Allowance = 0.0033 in.

	DIMENSIONS		
Hole	3.5000 $^{+0.0020}_{-0.0000}$	or	3.5000 3.5020
Shaft	3.5000 −0.0033		
	= 3.4967 $^{+0.0000}_{-0.0020}$	or	3.4967 3.4947

Example 2: ¾-in. shaft and hole, Class 7 fit. (see Fig. 2)

Shaft and hole tolerance = 0.0005 in.

Interference = 0.0004 in.

	DIMENSIONS		
Hole	0.7500 $^{+0.0005}_{-0.0000}$	or	0.7500 0.7505
Shaft	0.7504 $^{+0.0005}_{-0.0000}$	or	0.7504 0.7509

Example 3: 1.625-in. shaft and hole, Class 4 fit. (see Fig. 3)

Shaft tolerance = 0.0005 in.

Hole tolerance = 0.0007 in.

Allowance = 0.000

	DIMENSIONS		
Hole	1.6250 $^{+0.0007}_{-0.0000}$	or	1.6250 1.6257
Shaft	6.6250 $^{+0.0000}_{-0.0005}$	or	1.6245 1.6250

FIG. 1

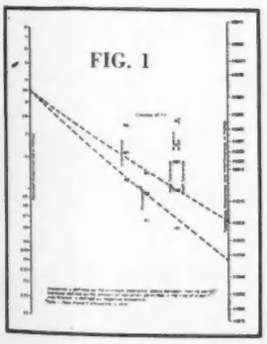


FIG. 2

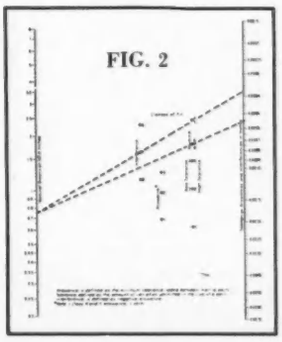
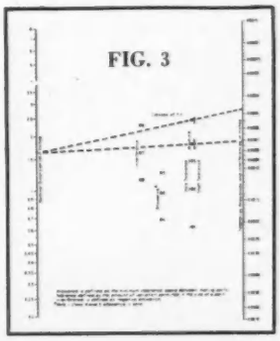


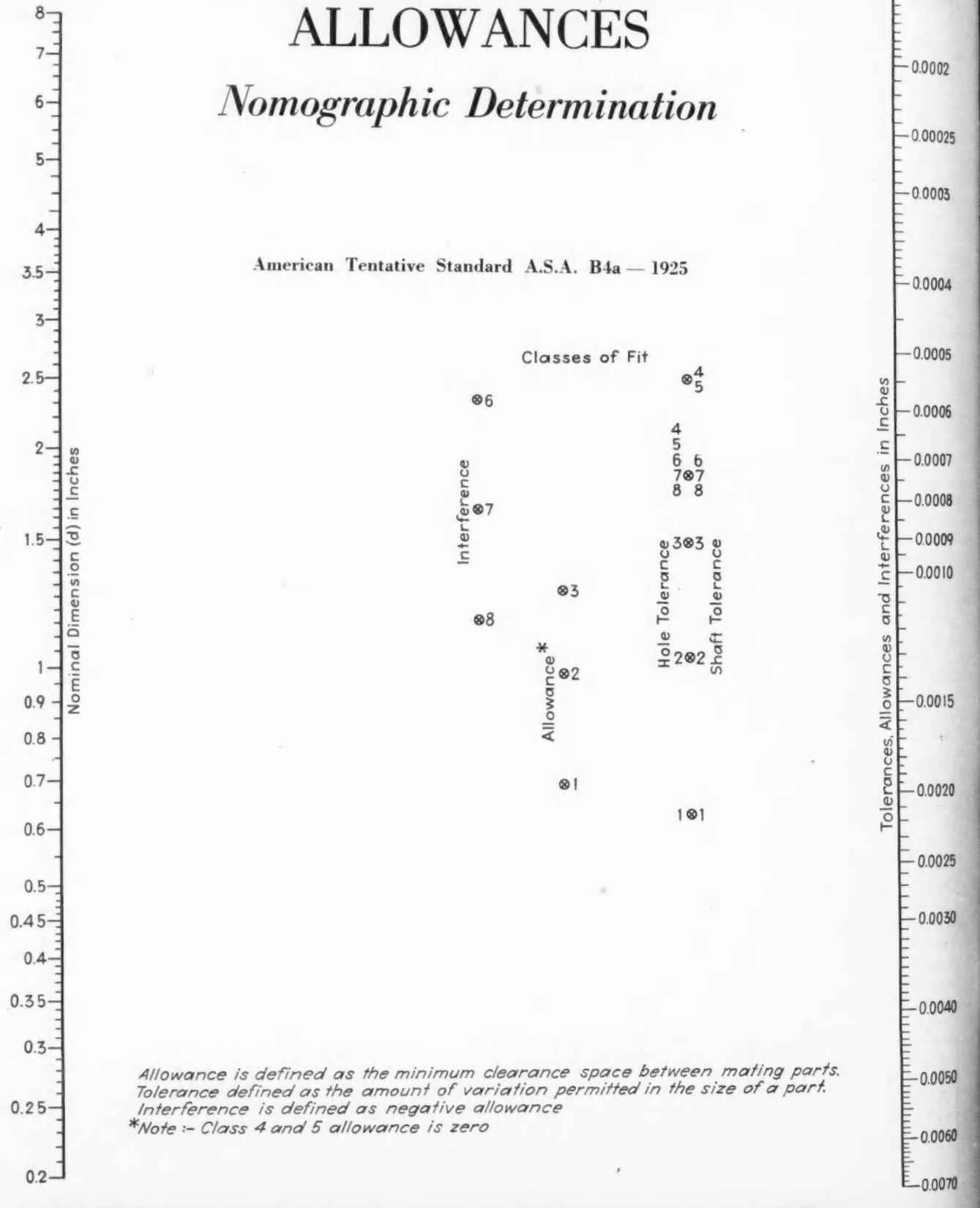
FIG. 3



TOLERANCES AND ALLOWANCES

Nomographic Determination

American Tentative Standard A.S.A. B4a — 1925



Allowance is defined as the minimum clearance space between mating parts.
Tolerance defined as the amount of variation permitted in the size of a part.
Interference is defined as negative allowance
**Note :– Class 4 and 5 allowance is zero*

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